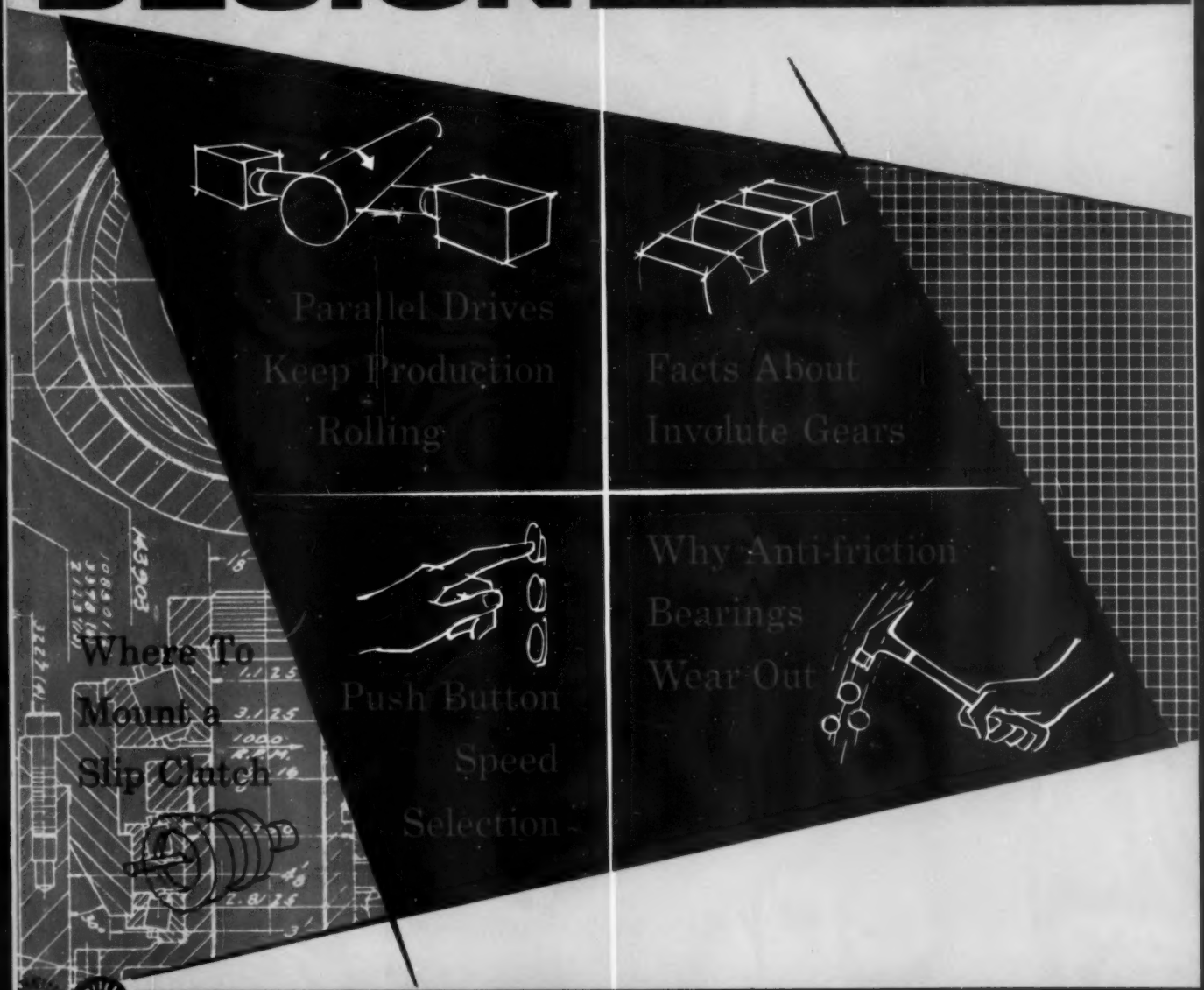
POWER TRANSMISSION DESIGN

FEBRUARY 1959

OEM · · MAINTENANCE



Parallel Drives Keep Production Rolling

Facts About Involute Gears

Push Button Speed Selection

Why Anti-friction Bearings Wear Out

Where To Mount a Slip Clutch

THE MAGAZINE OF MACHINE DRIVES

News and ideas for designers and plant engineers who use power drive equipment

Franklin INST-O-VERSE® Motors

ENCOURAGE FRESH THINKING FOR NEW PRODUCT APPLICATIONS WHERE GEAR REDUCTION IS NEEDED

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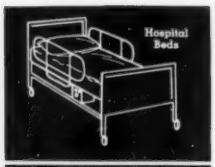
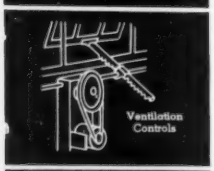
Ratios to your specifications through gear reduction such as epicyclic, worm, or bevel and other gears, PLUS instant, full-power reversibility extends power-package applications. These open new possibilities for competitively priced, high-reliability products. Hospital beds and garage doors are but two of today's newer products improved with these Franklin Motors.

AUTOMATIC OR MANUAL TRANSDUCERS ENERGIZE SINGLE OR DOUBLE POLE SWITCHES FOR ON, OFF, FORWARD OR REVERSE OPERATION OF ONE OR MORE OUTPUT SHAFTS.





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Franklin Electric Co., Inc.
HOME OF DEPENDABLE ELECTRIC MOTORS
BLUFFTON, INDIANA

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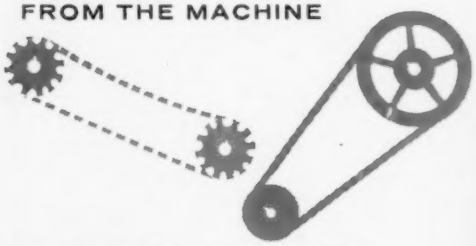
maurey DRIVES

get full power
OUT OF THE MOTOR

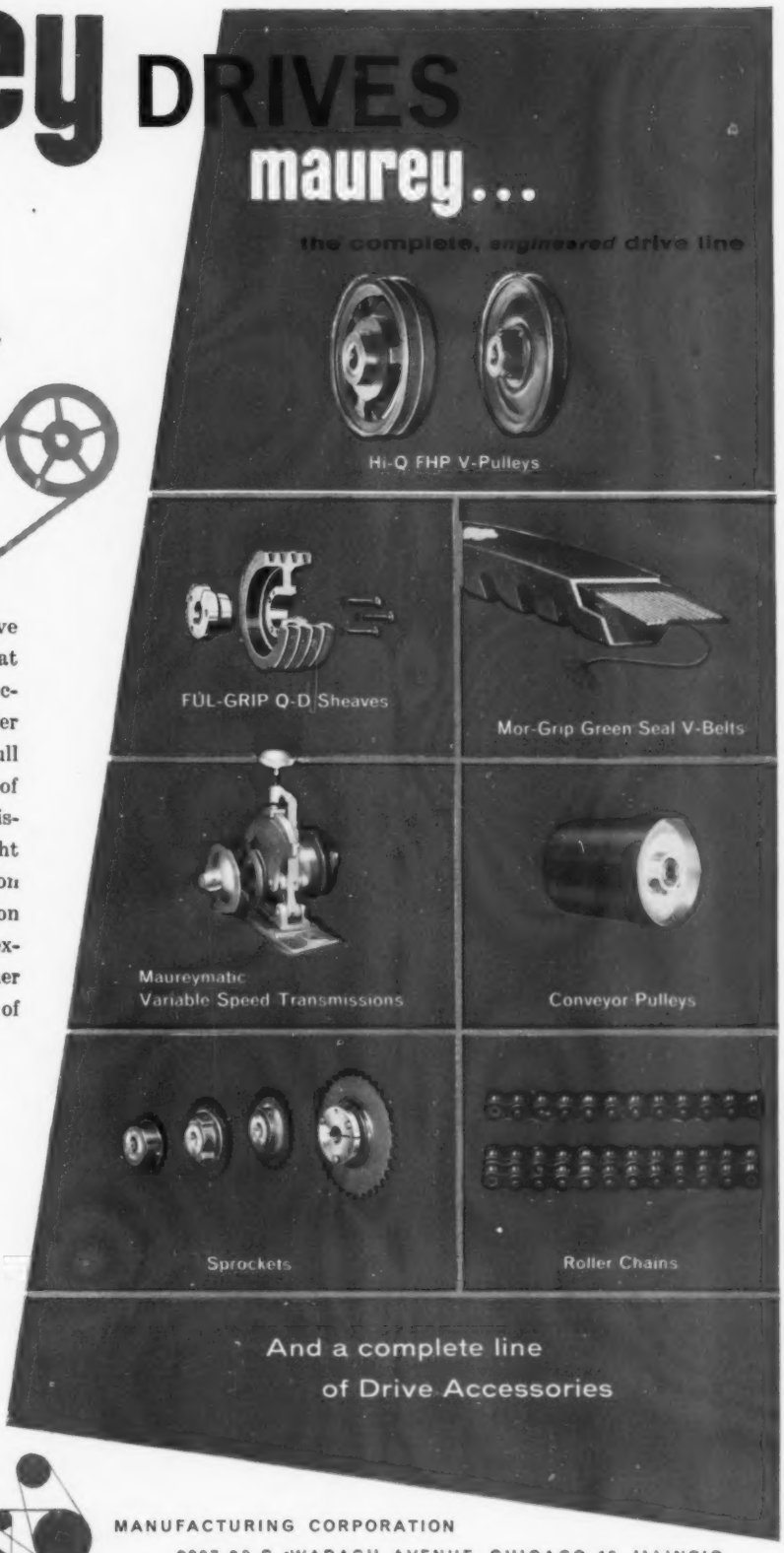
full production
FROM THE MACHINE



Before any machine puts out, a drive has to put power in, and that's what Maurey Drives do with maximum effectiveness. Maurey V-Drives and Roller Chain Drives are engineered to get full power from the motor to the point of production. A Maurey power transmission specialist will help you get the right drive on the job for almost any situation . . . the stop and go of a refrigeration drive, the smooth power flow of a textile drive, the pulsation of a rock crusher drive, or the round-the-clock action of an oil drilling drive.

Call your local Maurey Drive distributor, or write for any one or all of these Maurey Power Transmission Catalogs:

- FHP V-Drive Catalog F-10
- V-Belt Catalog No. 55
- Multiple V-Drive Catalog, MVD-58
- Maureymatic Variable Speed Transmission Catalog No. MM-58
- V-Drive Engineering Manual
- Roller Chain Drive Catalog D-58
- Conveyor Pulley Catalog CV-1



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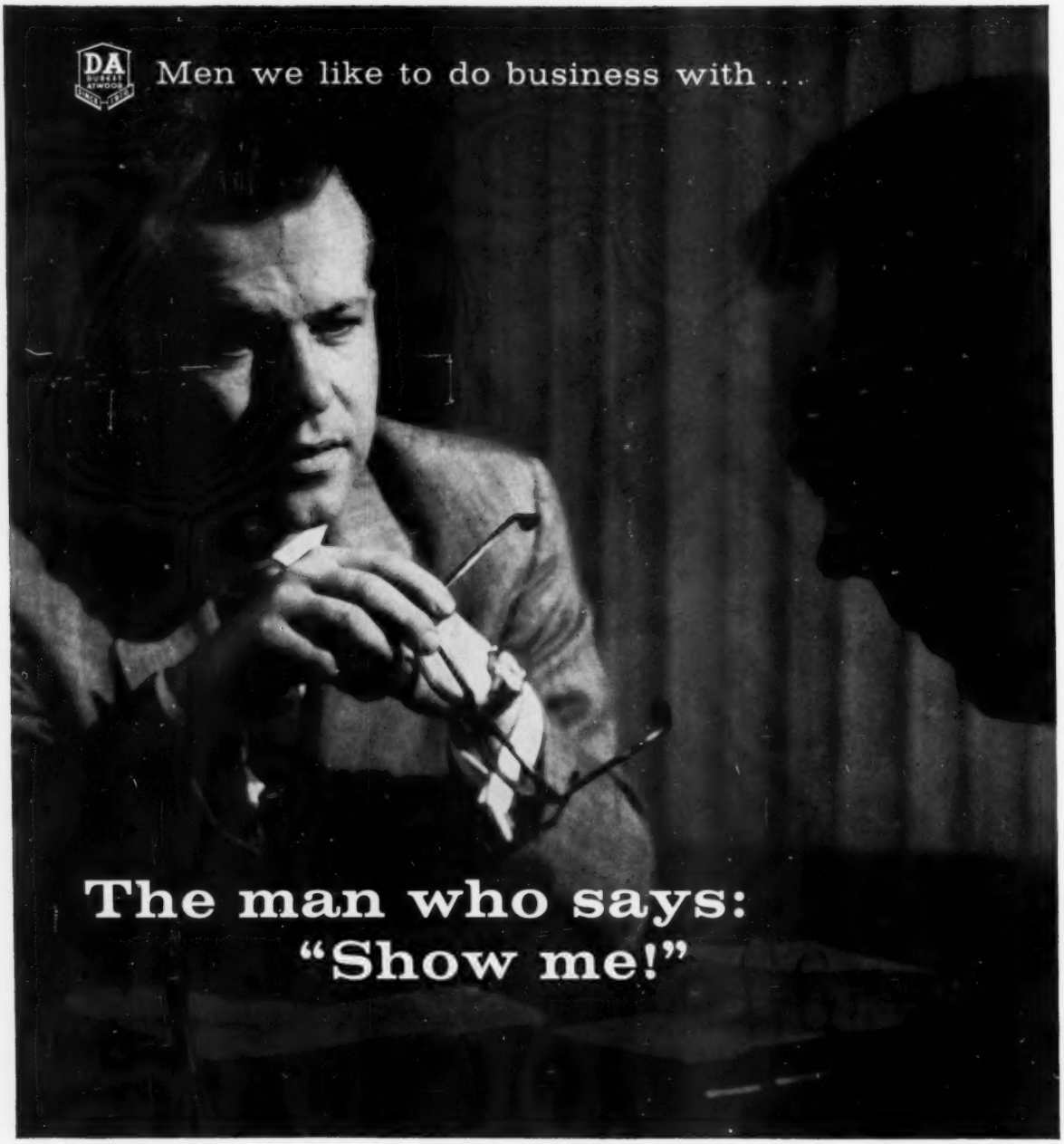

Every day we welcome the opportunity to show operating and maintenance engineers how DURKEE-ATWOOD V-Belts will meet their needs better than any other V-Belt manufactured. We prove this claim in many ways: We demonstrate *Quality* to meet demanding needs, quality that engineering and constant research have developed and severe usage has proven . . . We tell about our prompt *Service*. When you need special attention and help "right now," we're Johnny-on-the-spot . . . We explain fully how DURKEE-ATWOOD V-Belts provide real money *Savings* . . . DURKEE-ATWOOD, one of America's pioneer V-Belt manufacturers, offers a complete line of Multiple V-Belts, General Duty V-Belts and Steel Cable V-belts—built to *last!* We've got the belts to meet your needs exactly.

DURKEE-ATWOOD V-BELTS

DURKEE-ATWOOD COMPANY • MINNEAPOLIS 13, MINNESOTA

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READERS' GUIDE

February 1959, *volume 1 number 2*

FEATURE SECTION

15 **HOW TO DETERMINE SLIP CLUTCH POSITION** by Joseph Kaplan
You have several choices when it comes to mounting a slip clutch. Which choice will be yours depends on qualifying factors and specific applications. Read how to know them.

18 **PARALLEL DRIVES KEEP PRODUCTION ROLLING** by F. A. Depweg
Read how this firm keeps production going without interruptions. They use the "watch-dog" system of drives. Get the details on how one keeps guard while the other works.

20 **PUSH BUTTON SPEED SELECTION** by Albert Dall
Ever wonder how speed could be controlled by pressing a button? This unusual article shows how this dream went from the concrete-idea stage to become efficient reality!

23 **ELIMINATE THE GUESS WORK IN INVOLUTE SPUR GEARS** by Gatcomb & Prowell
Why are these used more than all other types of gears? Reasons are given in this detailed, informative article by two experts in the field.

IDEAS FOR DESIGNERS AND PLANT ENGINEERS

25 **TENSED WIRE ELIMINATES SHAFT AND BEARINGS**
One of the best ways to overcome mechanical wear and friction at armature supports is revealed for the first time. Find out how this was done. An unusual idea!

26 **ECCENTRIC MOTORS PRODUCE SINE WAVE FORCES**
Development of this system was a major advancement in the determination of response of accelerometers under certain conditions. The idea has proven itself.

26 **CONSTANT SPEED UNIVERSALS STABILIZE ROLL VELOCITY**
Clever idea-work solved the problem of speed variations produced by conventional universal joints in an aluminum rolling mill.

27 **CLUTCH BRAKE FITS SHORT SPACE NEEDS**
When working in tight space, keeping inside the circumscribed area is easy when the right idea is developed. Find out how weight went down and open-space increased.

28 **GEARMOTORS SYNCHRONIZE DIFFERENT WHEEL SPEEDS**
How to close moving leaves of an auditorium roof, do it smoothly, and do it in only two and one-half minutes—this was the problem. Engineers came up with an idea-solution.

29 **TIMING BELT STOPS VIBRATION**
Stopping vibration is very important, especially when efficiency of the entire production-operation is involved. A unique idea gave the answer to several drive-problems.

29 **FLAT BELT IMPROVES LOW SPEED EFFICIENCY**
There was one way to get better efficiency in the operation of a liquid oxygen compressor. It was discovered when an idea popped into the minds of engineers.

30 **TWISTED TIMING BELT SYNCHRONIZES THREE SPINDLES**
Eliminating chain drives and right-angle gears cut needed rpm in half. Read how this was worked out from an engineer's idea, and put to use.

30 **V-LINK BELT STOPS LONG SPAN STRETCH**
When you have a problem of stretch and looseness, how can it be solved? Here is an idea.

56 **OPEN DRIVE RESISTS MARINE CONTAMINATION**
Corrosive slime and raw water threatened open drive action. Read how contamination was fought off by the application of an engineer's idea.

BEARING SECTION

40 **WHY ANTI-FRICTION BEARINGS WEAR OUT** by Hudson T. Morton
What are the danger signals that forecast ruined bearings? They are listed here, in detail.

42 **PLAIN VS ROLLING CONTACT BEARINGS** by J. P. Vidosic
Find which type of bearing has least friction. Discover basic differences between bearings.

45 **LUBRICATED-FOR-LIFE VS CONVENTIONAL BEARINGS** by W. A. Williams
Do you know when to use one kind of bearing instead of the other? After reading this controversial article you can be sure of your choice and selection.

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FACTS

and figures on products and literature of interest to designers, engineers, and other power transmission design people can be had by circling the right numbers on the reader service cards found opposite page 48.

Browning

BEARING UNITS

Offer You 6 BIG Advantages

1 MALLEABLE IRON HOUSINGS
2 MORE COMPACT DESIGN
3 LIGHTER WEIGHT
4 SELF-ALIGNING BEARINGS
5 LIFETIME LUBRICATION
6 ECCENTRIC LOCKING RINGS

Browning combines all the cost-saving features of modern anti-friction bearings with a Browning exclusive—*unbreakable malleable housings*—to provide bearing units that are virtually indestructible. Because malleable is stronger than ordinary gray iron castings, Browning bearing units handle equal or heavier loads in smaller space and with less weight; operate longer with less downtime, lower labor costs.

It will pay you to investigate this quality line of rugged pillow blocks, flange blocks and take-up units. Write today for complete descriptive catalog BU-101.

Browning Manufacturing Company
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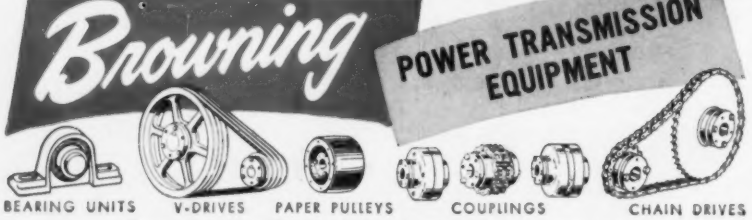
POWER TRANSMISSION EQUIPMENT



BEARING UNITS V-DRIVES PAPER PULLEYS COUPLINGS CHAIN DRIVES

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POWER TRANSMISSION DESIGN

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NEW Maxitorq 9000 SERIES*

ELECTRIC CLUTCHES or BRAKES

Featuring these

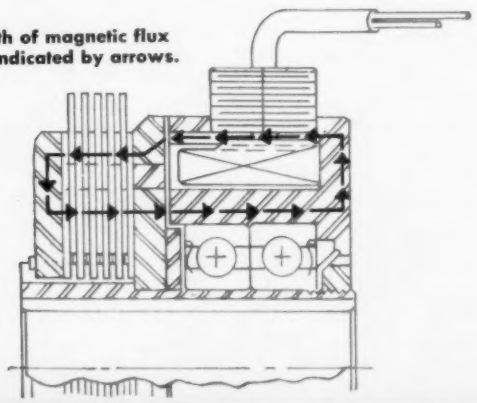
Advantages:

- **Compact and Simple Design**
- **Built to American Standards**
- **Stationary Coil**
- **No Slip Rings or Brushes**
- **Self-Adjusting—Easily Installed**
- **Drag-free Floating Neutral**
- **No Stressed Levers or Parts**
- **Use as Brake or Clutch**
- **Standard Flange Cups Available**

**Patent applied for.*

OPERATION INDUCED ENTIRELY BY *Magnetic Flux*

The new Maxitorq #9000 Series Electric Clutch is a compact, self-contained unit with operation dependent upon induced magnetic flux. As shown in the drawing below, this new Maxitorq Clutch consists essentially of a sealed magnetic coil and housing arrangement actuating multiple discs which also act as the flux path. Because the coil housing, supported on anti-friction bearings, is stationary, troublesome brush and slip ring construction is eliminated. When energized, the entire assembly consisting of pressure plate, inner and outer discs, and buttress plate is locked together by the magnetic force passing through it. When de-energized, Maxitorq separator springs instantly separate the discs creating air gaps which help to break up residual magnetism. Compactness, fast response, and flexibility of control make the #9000 Series clutch or brake ideally suited for a wide range of applications.



Path of magnetic flux is indicated by arrows.

Write, phone, or wire Dept. PT . now for Bulletin 90 describing the most advanced design of electric clutches or brakes available today.

THE CARLYLE JOHNSON MACHINE CO. • MANCHESTER, CONN.

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NEWS

from the power transmission field

(L to R) Falls, Farrell, Kingsley, Knudsen, Hancock

NEMA Names New Officers

ATLANTIC CITY, N. J.—New officers of the National Electrical Manufacturers Association and its many sections were elected at the 32nd Annual Convention.

Joseph L. Singleton, vice president, Industries Group, Allis-Chalmers Mfg. Co., was elected president. Vice presidents are: A. D. R. Fraser, president, Rome Cable Corp.; Frank H. Roby, executive vice president, Federal Pacific Electric Co.; N. J. MacDonald, president, Thomas & Betts Co.; Chris J. Witting, vice president, Westinghouse Electric Corp.; and W. R. Persons, president, Emerson Electric Mfg. Co. Treasurer is J. R. MacDonald, chairman of the board and president, General Cable Corp.

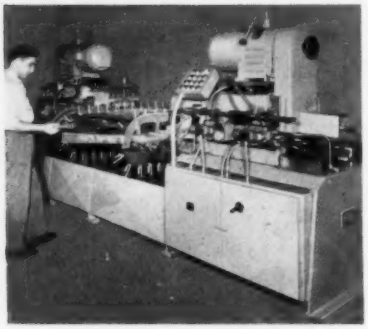
Members of the Motor and Generator Section elected at the meeting are: W. H. Kingsley, section chairman (sales manager Electrical Div., Fairbanks, Morse & Co.); M.S. Hancock, chairman, General Engineering Committee (assistant manager, Motor Engineering Department, Westinghouse Electric Corp.); C. W. Falls, vice chairman, General Engineering Committee (senior engineer, Motor and Generator Industry Standards, General Electric Co.); J. F. Farrell, chairman, Fractional Horsepower Subsection (sales manager, Industrial Motor Dept., Small Motor Div., Westinghouse Electric Corp.); and M. E. Knudsen, chairman, Integral Horsepower Subsection (motor sales manager, Motor and Control Div., Westinghouse Electric Corp.)

API discontinues viscosity standards

PHILADELPHIA, PA.—Standard viscosity oils for viscometer calibration will not be available from the American Petroleum Institute after January 1, 1959. ASTM now sponsors a program to provide these oils. By arrangement with the Cannon Instrument Co., State College, Pa., seven new standard oils for viscometer calibration are available.

ASTM tentative method D 445 and standard method D 88 for finding Saybolt viscosity will refer to the new standards. Prices are $15.00 per pint FOB State College, Pa.

V-belts drive new transfer machine



DETROIT, MICH. — Variable-speed V-belt drives power a new double-end transfer machine to perform as many as six forming operations on rounds or tube to 2 inches diam. A maximum of 3600 pieces per hour can be milled, bored, flared, cut off, press formed, etc. Machine is made by Walter P. Hill, Inc.

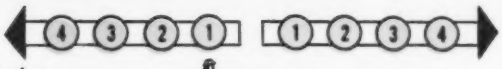
Multiple V-belts stop fan failure

CATASAUQUA, PA. — Oversize Multiple V-Belts provide continuous operation even though an individual belt may fail on a new line of induced-draft fans. Fans with motors of 5 hp or less use variable-pitch sheaves; larger fans have fixed drives and speed adjustments are made by changing sheaves. The fans are made by the Lehigh Fan & Blower Div. of the Fuller Co.

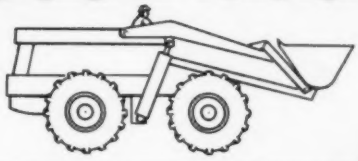
only the HYDRA-DRIVES® BDB

OFFERS ALL THESE MAJOR ADVANTAGES

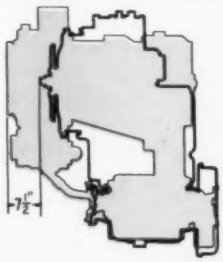
IN FULL-POWER SHIFT TRANSMISSIONS for equipment from 60 to 175 h.p.

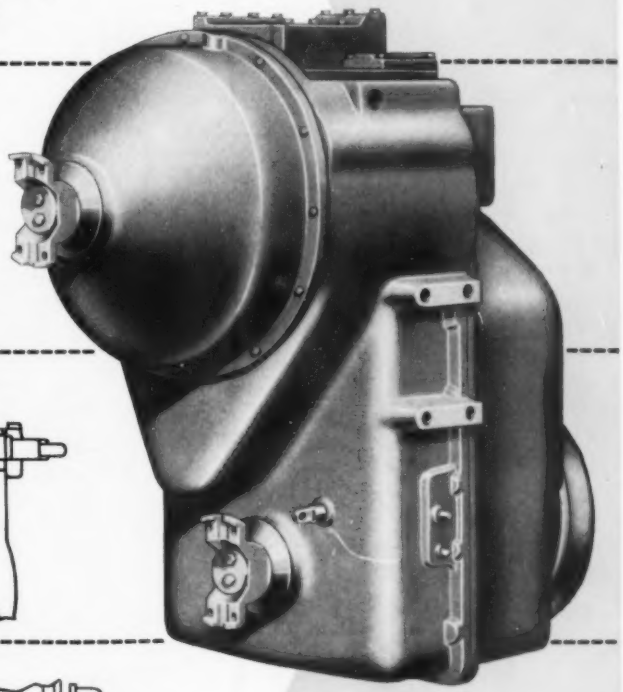


1. **4 speeds forward and reverse.** All power shifted! Provides maximum horsepower to load under all load conditions.

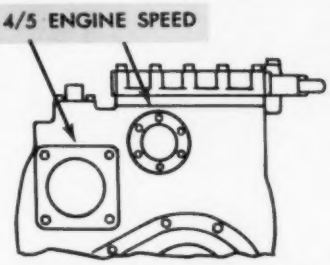


2. **Integral design.** Torque converter, transmission, oil passages, valving and oil sump are in *one* compact housing—7½" shorter than comparable models.

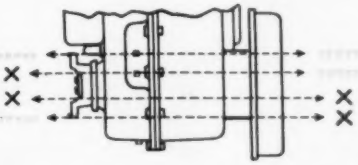




3. **Dual reduced speed pump drives.** Pumps can be driven at engine speed or ⅘ engine speed for longer life and increased horsepower to pump load. Single pump drive is also available.



4. **Full disconnect** provides four combinations of split drive . . . from torque on both shafts, to both shafts in disconnect.



SPECIALLY DESIGNED FOR SMALLER INSTALLATIONS

Rockwell-Standard's new model Hydra-Drives Full Power Shift Transmission is now available in sizes especially designed for smaller installations, such as front end loaders, fork trucks, scrapers, crane carriers, rubber tire tractors and military vehicles.

In addition, the Hydra-Drives BDB offers easier servicing and maintenance. There are fewer moving parts and bearings. The simple, rugged countershaft design and spur gears simplify maintenance.

HYDRA-DRIVES
ROCKWELL-STANDARD CORPORATION
TRANSMISSION AND AXLE DIVISION
DETROIT 32, MICHIGAN

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For more information circle No. 42 on the Reader Service Card

Silent chain drive powers two-stage compressor

Harrison, N.J.—A silent chain drive rated for 20,000 hours is used in a new "over-under" compressor. The new unit is built with the second stage compressor mounted on top of the first.

This added feature of the unit-design makes the compressor an item which is better for adaptation to the problems of modern compressor usage. Its general use in the field seems to be a certainty.

The unit has a clutch that can be engaged while the motor is running, permitting warm-up, with or without the compressor engaged. It eases strain on the engine, provides easier starts, less wear and tear. It is called model 365 Blue Brute. Worthington Corp.

Grinder and topping machines stop gear noise

Detroit, Mich.—Three major causes of noise in spiral bevel gears are eliminated with a grinder that relieves heel and toe ends of hardened gear teeth, and a tooth-topping machine that removes nicks and saw-tooth edges from teeth before heat-treat.

The driver-locator on the relief grinder revolves continuously at slow speed. The gear is placed on the locator with three pins that register with the gear mounting holes. No clamping is necessary. During grinding, the wheels rock in and out of contact with and follow the contour of the teeth. Time required to relieve all teeth on a 43-tooth ring gear is approximately 45 sec.

By reducing the active-noise of the gears, the general working atmosphere of the area where the machine is located shows a general improvement. Operator fatigue drops and the average worker-efficiency rises.

On the topping machine, the gear is placed on the driver locator and held by hand so that the tooth face bears against the driver. Both work and cutter rotate continuously while the teeth are topped. Time required to top a 10-tooth pinion is about 10 sec. Modern Industrial Engineering Co.

Precision forming process permits high production

Chicago, Ill.—A new process has been announced that successfully combines advantages of investment casting with other precision methods such as coining and cold forming. Known as the Minicast process, parts can be produced from almost all castable alloys and metals at production runs up to 50,000 pieces per day.

The process provides dimensional tolerances plus or minus 0.001 in; cross-sectional thickness as low as 0.020 in; superior surface finish of 30 to 50 microinches.

All types of products have been turned out with this process, including gears, sprockets and other power drive components. Casting Engineers, Inc.

Dallas distributor adds finish bore sprockets to line

Dallas, Texas—The Acme Chain Corporation revealed that the firm's new Dallas, Texas, warehouse now carries a complete line of finish-bore sprockets.

These include a total of 433 bores, all different, and all ready for use without reboring. The entire stock is ready for delivery.

Acme officials estimate that the Dallas warehouse stock is one of the most complete lines of finish-bore-sprockets in the southwest territory.

NLGI paper reprints are now available

Chicago, Ill.—The following papers are now available and can be had by writing to T.W.H. Miller, General Manager, 4638 J. C. Nichols Parkway, Kansas City, Mo. They were presented at the National Lubricating Grease Institute's regular annual meeting which was held in October, in conjunction with the American Gear Manufacturers' Association.

Yield Point of Lubricating Greases, by J. L. Dreher and D. W. Criddle, California Research Corp. (On page 56, this issue.)

Scientific Detection In Grease Lubricating Problems, by R. J. Ronan and M. C. McLaren, The Texas Co.

Milling of Grease by C. L. Johnson, Jesco Lubricants Co.

The Mechanisms of Dispersion, by K. H. Birkett, Battenfeld Grease & Oil Corp., Inc.

Practical Application of Milling, by J. J. Dickason, Jesco Lubricants Co.

A Preformed Organic Thickener, by J. E. Kline, Amoco Chemicals Corp., W. L. Hayne, and T. P. Traise, Standard Oil Co. of Ind.

Gear Design and Means of Lubrication, by J. H. Allen, Farrel Birmingham Co.

Modern Techniques for Spray Lubricating Industrial Gears by E. J. Gesdorf, The Farval Corp.

Industrial Gear Problems From The Lubricant Manufacturers Viewpoint, by G. H. Davis, Shell Oil Co.

Steel Mill Gear Lubrication Problems by A. E. Cichelli, Bethlehem Steel Corp.

Recent Developments in Industrial Gear Lubricants by E. S. Reynolds, Socony Mobil Oil Co., Inc.

Current and Future Military Requirements by R. J. Horwath, Military Petroleum Supply Agency.

Development of Extreme Pressure Greases by Dr. R. K. Smith, E. F. Houghton & Co.

Pressure and Temperature Effects on Flow Properties of Grease by Dr. H. Eyring and Dr. A. T. Ree, Univ. of Utah.

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MEN

of the power transmission industry

T. B. WOOD'S SONS CO. sales representatives. Front row, left to right: E. R. Kleinlein, H. D. Clark, Sr., G. W. Kuhn, W. T. Wonnacott, R. C. Reese, R. J. Ward. Second row, left to right: W. G. Johnson, T. C. English, R. S. Carlson, R. G. Cahill, C. O. Wood. Third row, left to right: B. W. Parsons, T. F. Lynch, R. S. Kries, R. C. Rotz, B. F. Watson. Fourth row, left to right: R. E. Lindley, H. M. Bowman, H. G. Tondreau, R. L. Garretson, G. M. Henderson, R. J. Tondreau, C. R. Shank, H. T. Kendall, and Joe Seawell, Jr.

T. B. Wood's Sons Co. creates four sales areas

Four regional sales managers now head four major marketing areas of the T. B. Wood's Sons Co., of Chambersburg, Pennsylvania.

Announced at the firm's tenth annual sales conference, the new officers are: R. L. Garretson, the northeastern region; R. G. Cahill, southwestern; Joe Seawell, Jr., southern; and Richard J. Ward, northern territory. Ward's headquarters will be in Chicago, Ill.

The conference, which resulted in new sales policies planned to make closer distributor contacts, was held in the company auditorium at Chambersburg.

Presiding were: C. O. Wood, Jr., president; and H. M. Bowman, vice president.

G. E. Promotes Two

New vice presidents of the General Electric Co., in New York, are Raymond W. Smith, and Harold A. Strickland, Jr.

Smith, General Manager of the Transformer Division, is in Pittsfield, Mass. Strickland is General Manager of the Electronics Division.

Smith joined GE in 1928. Later, named auditor, he was assigned to Brackenridge, Pa., returning to Schenectady in 1933.

He was named general manager of his department in 1952.

Strickland was appointed consultant of energy utilization in the Engineering Services Division, in New York City. He lives in Mount Kisco, New York.

New Distributor

The W. C. Ducomb Co., Detroit, Michigan, has been appointed a distributor of Rawson centrifugal clutches and clutch couplings, according to Charles F. Trapp, Jr., Formsprag Co. vice president.

The firm also distributes ball and roller bearings, power transmitting and industrial materials, and surface protection coatings.

PT, Inc., Appoints V.P.

BULLOCK

Harry Bullock became vice president of Power Transmission, Inc., in Denver, Colorado. The company specializes in bulk materials-handling, also crushing, screening, washing ,and conveying. Before joining Power Transmission, Inc., Bullock supervised major installations all over the world.

New SKF research men

Thomas W. Morrison is appointed assistant to the vice president in charge of research and engineering at SKF Industries, Inc., Philadelphia, Pa. He is a member of ASME, American Society for Metals, American Standards Association, Coordinating Research Council, and the Anti-Friction Bearing Manufacturers' Association.

Tilbor E. Tallian is now SKF Research Laboratory Manager. Frank W. Wellons is Chief Engineer. Wellons is chairman of the Lubrication and Wear Committee of the National Aeronautics and Space Administration (NASA).

West coast man for Barden

The Barden Corporation, makers of precision ball bearings, have assigned F. Brown McKee, product engineer, to their Los Angeles, California, offices. Herbert D. Williams, formerly product engineer in the Los Angeles office, has returned to the company's home-office engineering department in Danbury, Connecticut.

This ends a term of service in the western office for Williams, who has been stationed there since Barden opened the California offices approximately a year and one-half ago in July, 1957.

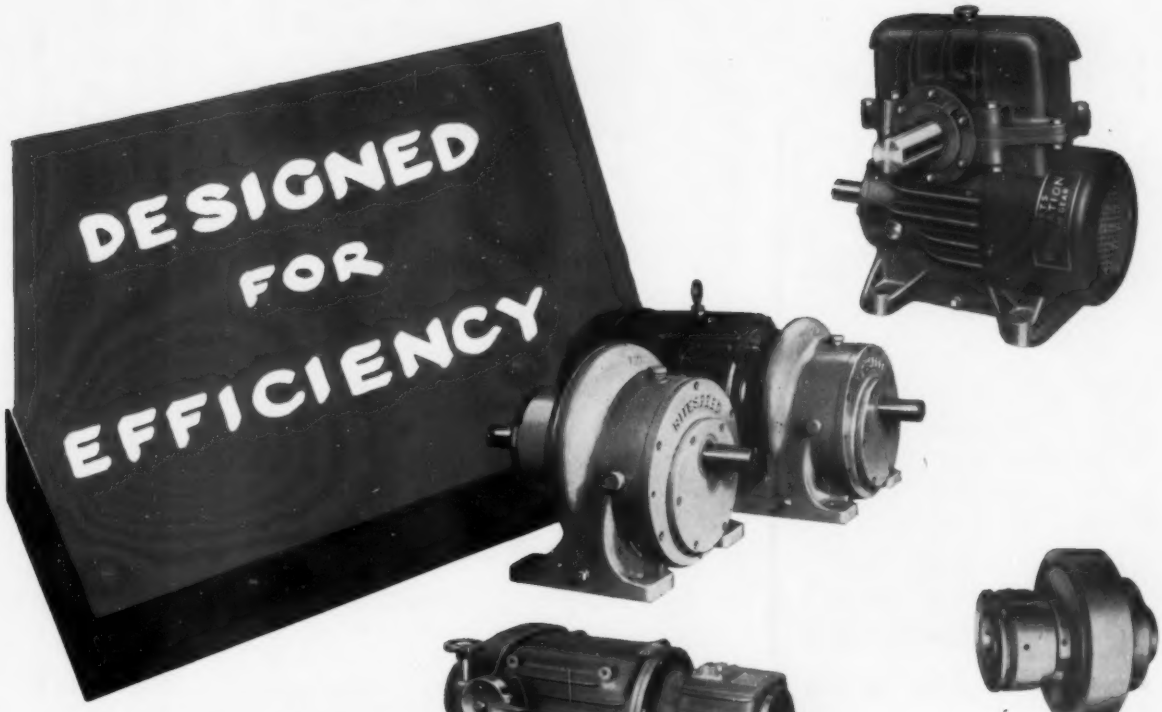
Cullman Wheel ups three

KYLANDS

Cullman Wheel Co., Chicago, Ill., announces the following promotions: George L. Rylands, Executive Vice President; C. Casey, Secretary of the company; and Walter R. Whittle assistant to the president while continuing his duties as Director of Advertising and Marketing.

New Marlin-Rockwell VP's

Bernard J. Shallow is now Vice President in Charge of Sales at the Marlin-Rockwell Corp., Jamestown, New York. Franklin C. Clark is newly appointed to Vice President in Charge of Human Relations. Shallow was formerly general sales manager, Clark was director of human relations.



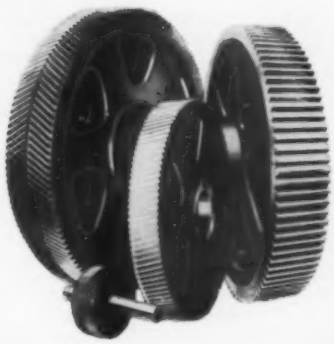
. . . AT LOW COST

Best quality, at lowest possible price is not just sales talk, it's a fact—when you buy at CROFTS.

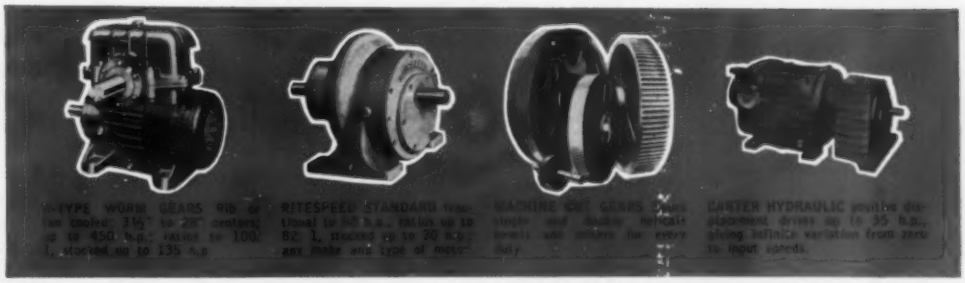
Over 70 years experience in the design and application of power transmission equipment stands behind CROFTS products.

Famous throughout the rest of the world for nearly three-quarters of a century, CROFTS are now able to offer for the first time in the States their full range of power transmission equipment.

Replacements available from stock . . . if needed. But the chances are it'll be a long time before you'll need to replace anything made by CROFTS.



For descriptive literature on the products shown here . . .



. . . and for the name of your nearest distributor, write to:

CROFTS U.S.A., INC.

2542 W. PETERSON AVE.
CHICAGO 45, ILLINOIS

EASTERN STATES DISTRIBUTORS
ROCKWOOD PULLEY MFG. CO., INC.
20 CROSBY ST., NEW YORK 13, N. Y.

MAKERS OF: Ritespeed Gearmotors • Ritespeed Reduction Gears • Worm Gearmotors • Worm Gear Reducers • Helical Gear Reducers • Spur and Bevel Gears • Motor Driven Rollers • Spiral Bevel Gear Units • Hydraulic Couplings • Friction Clutches • Carter Variable Speed Hydraulic Gears • Disc Type Friction Clutches For Industrial and Machine Tool Applications . . . . . .

For more information circle No. 37 on the Reader Service Card

Announcing-

THE NEW
SIMPLEX

A 3/16" or larger O.D. Universal Joint featuring:

- **Minimum Static Torque Rating of 250 inch-ounces**
- **Non-magnetic stainless steel forks and bronze ball**
- **New simple 3-part design**
- **Minimum back lash**

Newest addition to the Curtis line is the Simplex, designed to fill the need for a small-size universal joint with a high Static Torque Rating. The Simplex is available in 3/16" and larger outside diameter. Made of non-magnetic material and incorporating a new design inherently strong in torque, the Simplex is particularly well adapted to electronic instrumentation.

SIMPLEX SPECIFICATIONS

Catalog Number	S3	S3B	S7	S7B
Static Torque Rating	250 Inch-Ozs.	250 Inch-Ozs.	200 Inch-Lbs.	200 Inch-Lbs.
O.D.	3/16"	3/16"	7/16"	7/16"
Bore	None	3/32" Dia. 5/16" Deep	None	7/32" Dia. 9/16" Deep
Total Length	1"	1"	2"	2"
Max. Angle of Operation	20°	20°	20°	20°

CURTIS TRADE MARK
UNIVERSAL JOINT CO., INC.
22 Birnie Ave., Springfield, Mass.
As near to you as your telephone

Circle No. 11 on the Reader Service Card

EDITOR'S PAGE

Already in preparation are important articles of interest for PTD's readers. Read through this preview . . .

David R. Cartwright
editor

What about next month For Power Transmission Design readers?

Ever try graphite . . .

. . . as a bearing lubricant? Did you ever check its lubricating qualities?

Strangely enough, graphite has no intrinsic lubricating qualities itself but depends on adsorbed surface film for reducing friction. It also depends on the carrier in which it is motivated into the bearing area and is one of the few lubricants that can work at high temperatures (100F to 1000F) away above temperatures that break down oil and grease.

See more about graphite and carriers next month in POWER TRANSMISSION DESIGN.

Ever consider cost . . .

. . . when selecting gear reducers? Did you know there are several hundred dollars difference between equivalent reducers of given capacity?

Next month, Bill Williams, consulting editor and regular contributor to POWER TRANSMISSION DESIGN, touches on this subject with great impact. He compares various types of commercially-available right angle reducers from an initial cost standpoint. Don't miss it.

Small servo motors . . .

. . . can do a man-size job for you. See how a motor that develops only 100 in-lb torque can move two large external surfaces on an airplane with ease. See how the design engineer in charge of the project used steel cable wound around a quarter-section "quadrant" to position these surfaces with precise mechanical advantage.

See it in application next month.

Shaved gears . . .

. . . may be the answer to eliminating your gear-train problems whether on large or small applications. This also next month in POWER TRANSMISSION DESIGN.

Is a solid shaft . . .

. . . always necessary? Maybe you never considered using a hollow shaft on motors, gear boxes and other drive equipment.

Consider the simple analogy of a long, solid steel beam spanned between two buildings with a great sag in the center, versus an I-beam of the same length with no sag. You'll find that torsional loads can also be compared this way and can be carried equally as well with either hollow or solid shafts. Don't miss it next month.

These and more . . .

. . . ideas will come your way in the March issue. You can help by telling us your power transmission problems, whether they be mechanical, electrical, hydraulic or other, so that we can find the solution for you and all other readers.

You can also help by passing along information on how you successfully licked a problem. It might help someone else. Our feature pages are for men like yourself who have something to say . . . and say it.

NEW PRODUCTS

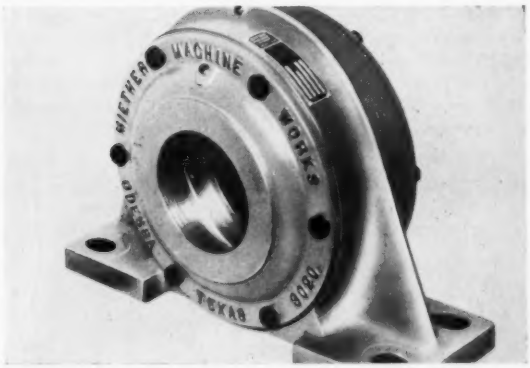
Speed changer for constant torque applications

Four-position motor-mounting and two-position gear-mounting make the speed changer adaptable to varying space conditions. Made up of one-piece motor, with driver-sheave shaft-ends supported between bearings—shaft deflections are reduced, efficiency goes up. Other speed-changer features: low weight per horsepower, small size, "in service" lubrication. Stepless speed changes come by rotating hand-wheel which increases and decreases pitch diameter of the sheave. Each unit can be assembled to give intermediate ratios of 5:1, 4:1, 3:1, 2:1, using 1800 or 1200 rpm.

Allis-Chalmers Mfg. Co., Milwaukee, Wisconsin.

Circle No. 246 on the Reader Service Card

20 housings accomodate 94 separate bearings



Interchangeable end caps on a new line of pillow blocks make it possible to accommodate 94 separate bearings with only 20 basic housings. Some can take eight different bearing sizes, 11 shaft diameters. The pillow blocks also accommodate spherical roller bearings. Housings are center-grooved for universal center lubrication. End caps have connections for side-lubrication. Different size bearings are installed by lifting end caps on the housing. Both end caps are removable for bearing installation or adjustment. Either labyrinth closures or oil seals are available on most caps, making them adaptable items.

Meither Machine Works, Odessa, Texas.

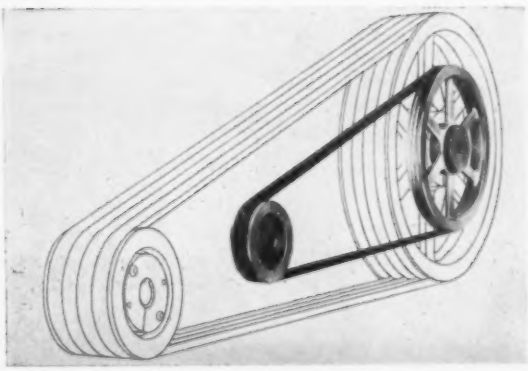
Circle No. 247 on the Reader Service Card

New V-belt cuts five cross sections to three

Provides complete coverage of the entire range of drive requirements with only three cross sections instead of the five formerly necessary. The high capacity belts and narrower sheaves make up a 4-dimensional advance in multiple V-belt drive design, that consists of reduced: belt and sheave width; sheave diameter; center-distance; and, overall drive-cost. Arched top of the V-belt prevents any distortion of the tensile section and allows each of the tensile cords to carry its share of the load. Concave sidewalls reduce wear on belt-sides, a cost-saving factor as far as necessary belt-replacement is concerned.

The Gates Rubber Co., Denver, Colorado.

Circle No. 248 on the Reader Service Card



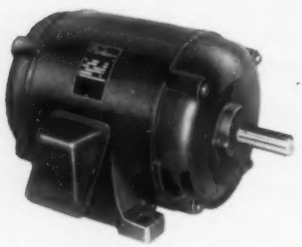
PEERLESS SPECIAL AND STANDARD ELECTRIC MOTORS

Peerless specializes in customer-manufacturer teamwork engineering of special motors. Unusual operating conditions and duty requirements are everyday problems to us. We are the exclusive motor supplier to many original equipment manufacturers. Our knowledge works to your advantage in reducing the engineering time required to solve your motor problem.

TYPES and RATINGS

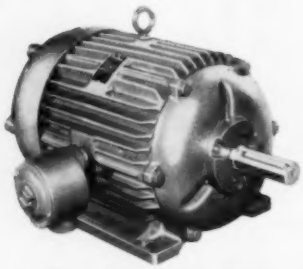
Single Phase	½ thru 10 HP
Polyphase	⅓ thru 30 HP
Direct Current	¼ thru 3 HP
TEFC (single phase)	1 thru 10 HP
TEFC (polyphase)	1 thru 30 HP
Explosion-Proof	Same as TEFC
Torque	2 oz. in. to 200# ft.

Special Designs to Specifications



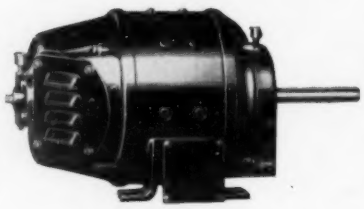
DRIP-PROOF (open type)

Completely protected against dripping liquids and falling particles. Surfaces are smooth and symmetrical. Frame sizes from 56 to 324 as shown. Frames 324 and 326 have welded frame. Cast-iron construction. Sleeve or ball bearings.



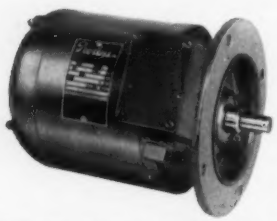
TEFC and EXPLOSION-PROOF

External fan draws cool air across motor toward driven machine. Heat from machine is not drawn across motor. Explosion-proof designs approved for Class I and II duty. Special mounting designed to meet application requirements. For HP ratings see table above.

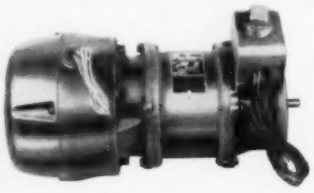


DIRECT CURRENT

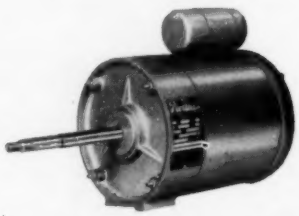
Designed for rough usage. High starting torque, good overload capacity and high electrical efficiency. Shunt, series or compound windings. Constant speed generators in ratings from 150 to 3000 watts. See HP ratings above.



WEATHER-TIGHT SPECIAL FLANGE



EXPLOSION-PROOF TORQUE MOTOR WITH BRAKE



SPECIAL FLANGE REVERSING HOIST MOTOR, SINGLE PHASE

ENGINEERING DATA—These bulletins are available from Peerless: Space-Saver, SP-1; Torque, T-1; and SDA-155 describing the complete line. Peerless builds many special mountings. Various modifications—special shaft features; paint and varnish treatments; Class A, B, F, H insulation, etc. are available. Peerless builds to the standards and specifications of JIC, AIEE (including AIEE No. 45 Marine Duty), ABS, Federal and Military.

ELECTRIC MOTOR DIVISION

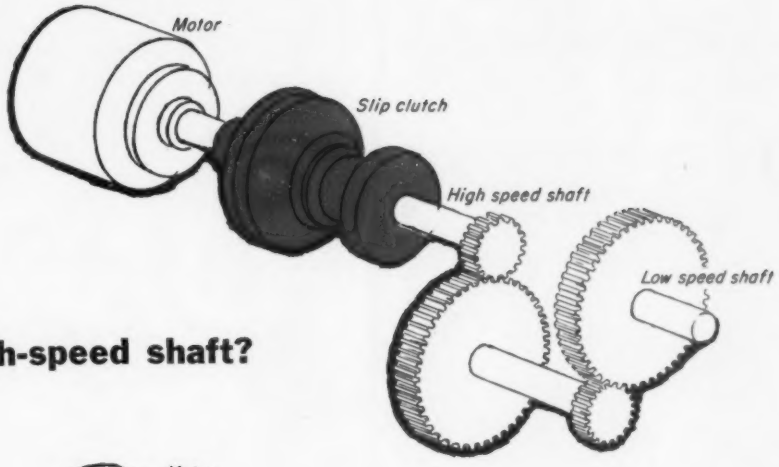
THE *Peerless-Electric*® COMPANY

FANS · BLOWERS · MOTORS

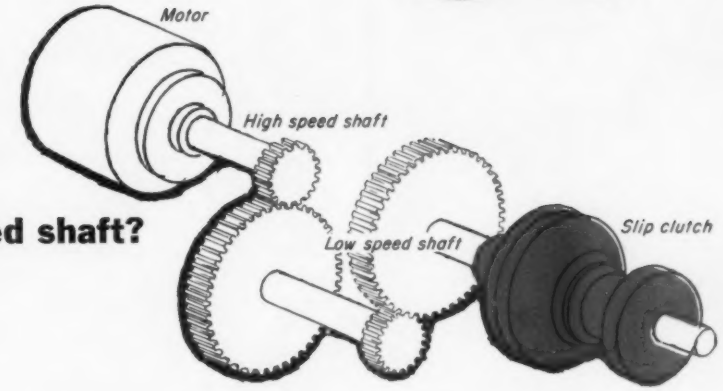
1512 W. MARKET ST. • WARREN, OHIO

For more information circle No. 39 on the Reader Service Card

For overload problems, do you:



Mount it on the high-speed shaft?



...or on the low-speed shaft?

How to determine slip clutch position

By **JOSEPH KAPLAN**, *senior project engineer, Fairchild Camera and Instrument Corp.*

THE JOB OF A SLIP CLUTCH is to act as a coupling between two shafts, permit cyclic joining of the stationary to the rotating shaft and provide relative motion or slip at rated torque. It is this slip ability that regulates the acceleration of the stationary to the velocity of the rotating shaft.

You have several choices for mounting. One is to place it on the high speed shaft. This permits the use of a low torque capacity clutch. The other is to place it on the low speed shaft, which would mean using a higher torque clutch. The location greatly affects performance of the whole machine. Thus, the wise choice will give the best cycling rate at lowest power consumption as well as produce the least power dissipation within the clutch. Performance factors can be translated into:

1. Higher cycling rate means higher productivity.
2. Less power consumption means smaller motors.
3. Less power dissipation means less clutch wear.

Assumption

Assume that the slip torque capacity of a clutch is constant regardless of the relative velocity between the two shafts. In practice, slip clutches have different torque capacities at high and low slip velocities due to coefficient of friction variations. Thus, many clutches have a higher breakaway torque which drops appreciably once relative slip has been effected.

Assume also that no drop in the velocity of the rotating member is effected during acceleration. Generally, most designs for high speed machines tend to minimize this drop. However, the assumptions will permit accurate analysis anyway.

Example

Let us select a stationary shaft that must be accelerated cyclically to 600 rpm through a slip clutch. The stationary shaft has a friction load of 6 inch-lb. A full cycle consists of one revolution of the 600 rpm shaft and a dwell period of 0.025 seconds. A 12,000 rpm motor rotating continuously is the prime mover. A choice must be made between two slip clutches. One clutch, to be located on the 12,000 rpm motor shaft, has a rating of 0.5 lb-in of torque and a moment of inertia of the stationary coupling element of 10^{-5} in-lb-sec.^{2}.

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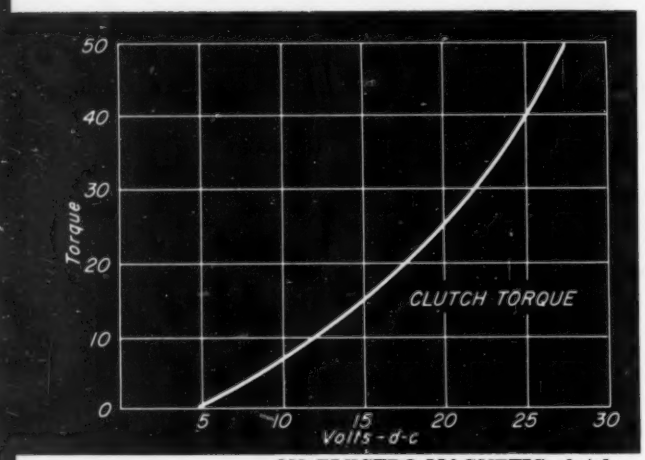


ON ELECTRO-MAGNETIC clutches, torque increases with voltage.

The other slip clutch, to be located on the 600 rpm shaft, has a rating of 10 lb-in and a moment of inertia of the stationary coupling element of 2×10^{-4} in-lb-sec.^{2}. The equivalent moment of inertia on the 12,000 rpm shaft to be accelerated by the first clutch is 5×10^{-6} in-lb-sec.^{2}. The equivalent moment of inertia on the 600 rpm shaft to be accelerated by the second clutch is 5×10^{-4} in-lb-sec.^{2}.

What is the maximum cycling rate that the transmission is capable of with either of the clutches?

What is the power, at the maximum cycling rates, to be supplied by the motor?

What is the power dissipated by each of the two clutches at the maximum cycling rate?

Solution

The duration of one cycle is 0.025 seconds plus acceleration, constant speed and deceleration periods required to displace one revolution of the 600 rpm shaft.

With High Speed Clutch

Using the 0.5 in-lb clutch:

Acceleration time

$$t_1 = w_a \frac{(I_c + I_s)}{(T_a - T_L) \times R} = 0.094 \text{ sec}$$

Deceleration time

$$t_2 = \frac{w_a \times I_s}{T_L \times R} = 0.021 \text{ sec}$$

Displacement of 600 rpm shaft during acceleration

$\theta_1 = \frac{1}{2} w_b t_1 = 2.95$ rad

Displacement of 600 rpm shaft during deceleration

$\theta_2 = \frac{1}{2} w_b t_2 = 0.66$ rad

Displacement of 600 rpm shaft during constant velocity

$\theta_3 = 2\pi - (2.95 + .66) = 2.67$ rad

Duration of constant velocity period

$$t_3 = \frac{\theta_3}{w_b} = 0.045 \text{ sec}$$

Duration of one cycle

$t_4 = t_1 + t_2 + t_3 + .025 = 0.185$ sec

Maximum cycling rate

$$C_a = \frac{1}{t_4} = 5.4 \text{ cps}$$

Power requirements at maximum cycling rate

$$P_a = \frac{T_L W_b t_3 + T_a W_a t_1}{t_4} = 394 \frac{\text{in-lb}}{\text{sec}}$$

Power dissipation in the slip clutch

$$P_c = \frac{T_a \theta_1}{R t_4} = 159 \frac{\text{in-lb}}{\text{sec}}$$

Low Speed Clutch

Using the 10 in-lb clutch:

Acceleration time

$$t_5 = \frac{w_b (J_c + J_s)}{T_b - T_L} = 0.011 \text{ sec}$$

Deceleration time

$$t_6 = \frac{w_b \times J_s}{T_L} = .0052 \text{ sec}$$

Displacement of 600 rpm shaft during acceleration

$\theta_4 = \frac{1}{2} w_b t_5 = 0.34$ rad

Displacement of 600 rpm shaft during deceleration

$\theta_5 = \frac{1}{2} w_b t_6 = 0.16$ rad

Displacement of 600 rpm shaft during constant velocity

$\theta_6 = 2\pi - (\theta_4 + \theta_5) = 5.78$ rad

Duration of constant velocity period

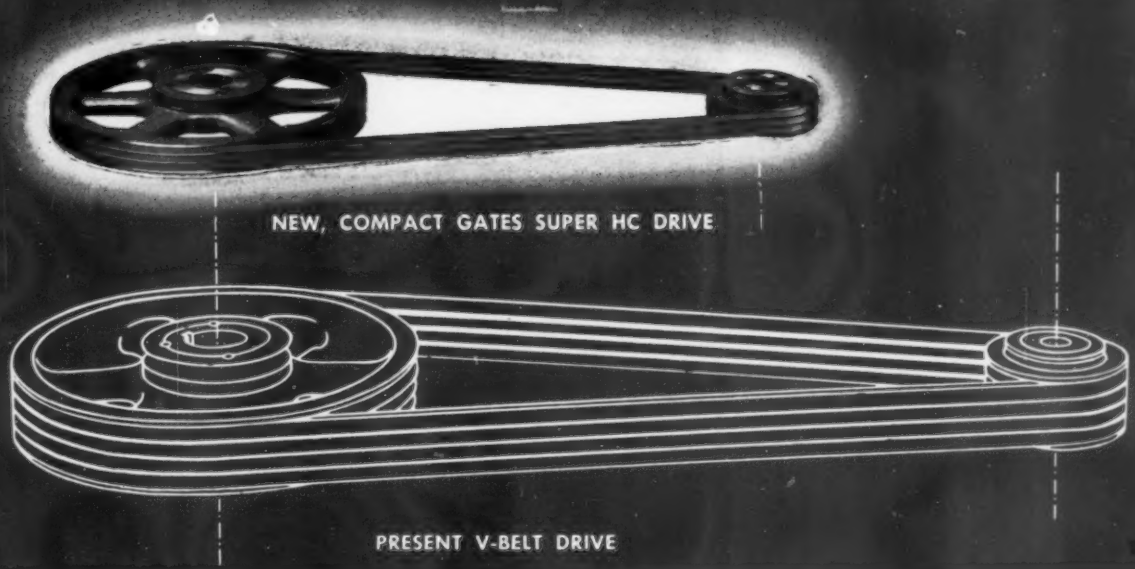
$$t_7 = \frac{\theta_6}{w_b} = 0.092 \text{ sec}$$

Continued on page 56

Nomenclature

I_c = moment of inertia of stationary element of high speed clutch, in-lb-sec^2
I_s = equivalent moment of inertia of stationary elements of high speed shaft, in-lb-sec^2
J_c = moment of inertia of stationary element of low speed clutch in-lb-sec^2
J_s = equivalent moment of inertia of stationary elements of low speed shaft, in-lb-sec^2
P = power requirement in-lb/sec
P_c = power dissipation, in-lb/sec
R = transmission ratio = $\frac{600}{12{,}000} = \frac{1}{20}$
T_a = torque capacity of high speed clutch, lb-in
T_b = torque capacity of low speed clutch, lb-in
T_L = load torque, lb-in
t = duration of period, sec
γ = angular acceleration, rad/sec^2
θ = angular displacement, rad
w_a = angular velocity of rotating element of high speed clutch, $\frac{\text{rad}}{\text{sec}}$
w_b = angular velocity of rotating element of low speed clutch, $\frac{\text{rad}}{\text{sec}}$

New high capacity V-belt revolutionizes drive design



Makes drives far more compact... cuts cost as much as 20%

Here is a major advance in the field of power transmission — the fully proved Gates Super HC V-Belt, developed in the world's largest belt-testing laboratories at The Gates Rubber Company.

With the Gates Super HC V-Belt, you can have the lowest-cost, lightest-weight, most compact multiple V-belt drive you can put on any machine! Sheave diameters can be reduced up to 50%, sheave widths 30% to 50%, center distances 20% and more!

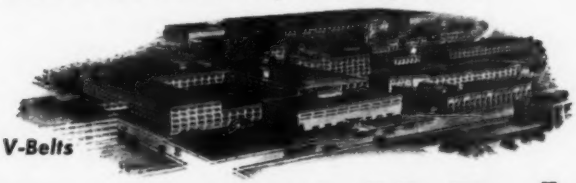
On new drives, the cost of a Gates Super HC V-Belt Drive is as much as 20% less than present V-belt drives of the same hp capacity!

Learn More About The Cost-Saving Super HC Drive

For detailed information on the Super HC Drive, contact your nearby Gates distributor or Gates Office for new booklet, "*The Modern Way to Design Multiple V-Belt Drives.*" For utmost space, weight and dollar savings in new drives or drive replacements, specify Gates Super HC V-Belts and Sheaves. Stocks immediately available in principal industrial centers.

The Gates Rubber Company Denver, Colorado
Gates Rubber of Canada Ltd., Brantford, Ontario

Gates
World's Largest Maker of V-Belts



Gates Super HC V-Belt Drives

For more information circle No. 22 on the Reader Service Card

If one drive unit goes out, the others will take the load. Motors, couplings, gear-boxes or clutches can be taken out of service but . . .

THREE-QUARTER MILE-LONG plate-glass line stays on continuous service despite breakdowns at Ford Motor Co., Nashville, Tenn.

Parallel drives keep production rolling

By **F. A. DEPWEG**, *manager, General Machinery, Hamilton Division, Baldwin-Lima-Hamilton Corp.*

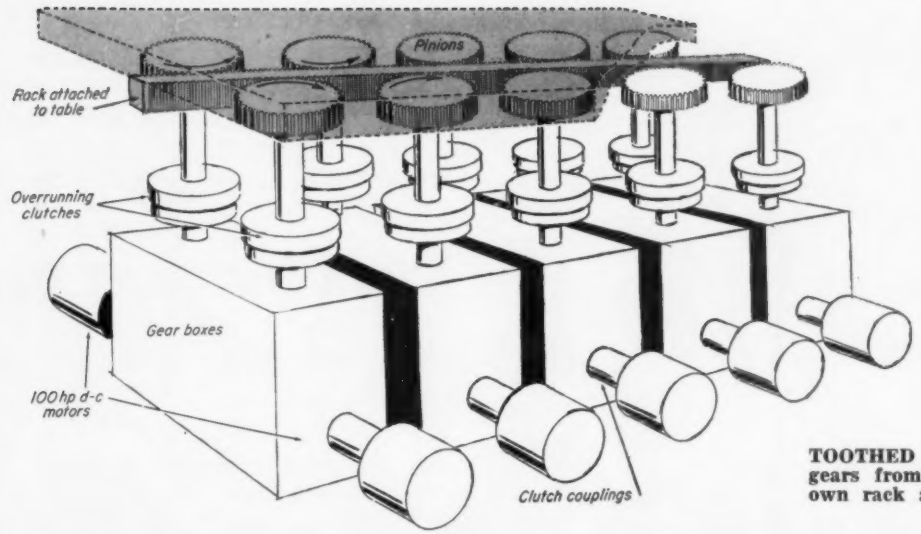
A NEW PLATE GLASS grinding and polishing line stays on continuous service—24 hours a day, five days a week. To do this, a parallel drive system is used so that if any part of either system has to be shut down, the remaining units keep production going.

There are two of everything in the drive system. There are two shafts rising from the top of each gear box. Each gear box has a twin set of reduction gears enclosed independently of each other. There are two banks of 100-hp motors on opposite sides of the gear boxes; and the entire drive system is located in a pit below the production floor.

The drive shafts power the pinions which impart forward motion to toothed racks attached underneath each table. The racks are toothed on both sides. Each table pushes the table ahead of it. When all drive units are operating, all pinion gears are meshed with the racks on the tables above.

The "line" actually consists of two lines joined to form an oval. Each of the two lines has 32 grinding and 50 polishing stations. Because the two lines are joined at either end by semi-circular turn-around section, the installation has the appearance of being one circular operation. However, each line is a complete and self-contained operation.

Cast-iron tables, each weighing 17 tons, form two



TOOTHED RACK is driven by pinion gears from both sides. Each table has own rack attached underneath.

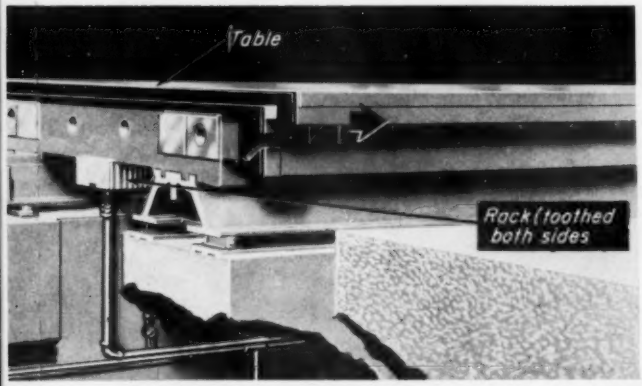
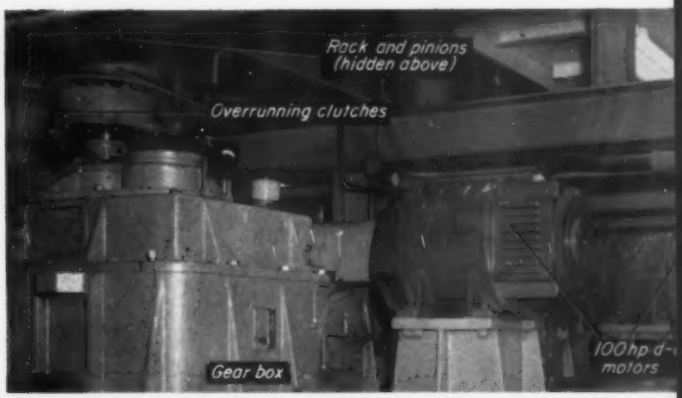


PLATE GLASS is mounted in plaster on tables which push one another forward.



OVER-RUNNING CLUTCHES allow any drive unit to be stopped without dragging on the system.

continuous ribbons, each 1,775 feet long as they are propelled down the lines in opposite directions. Fixed to each table by a bed of plaster is a section of glass measuring approximately 10 x 14 feet which is subsequently ground and polished to 1/8 or 1/4 in. thick. The glass passes under 82 grinding and polishing heads as it is forced down a line.

The tables ride on mating male bedways. There are always 124 tables on each line. This adds up to a ribbon of tables weighing 1,860 tons plus the weight of glass and plaster.

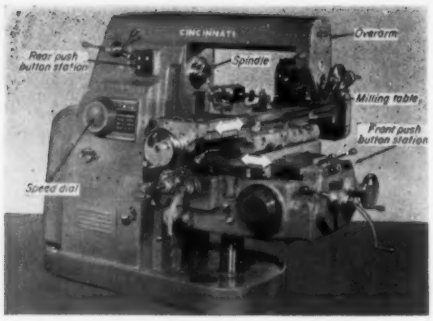
The power transmission system permits changes in load, provides easy maintenance, and operates at constant speed.

The drive pits are located under the starting section of each line. Each pit contains five drive units. Each unit consists of two 100-hp d-c motors, two high-speed over-running clutch couplings, a double gear box, two low-speed over-running clutch couplings and a pair of vertical pinions. As many as five or as few as two drive units or three motors can be operated to supply power to a line.

Power is transmitted from each motor through the gear boxes to high-speed over-running clutch couplings.

The coupling consists basically of concentric inner and outer races with sprags in the space between. When the motor is operating, the sprags by their wedging action, engage the inner and outer races to transmit power. They provide torque of 3,390 foot/pounds and engage at an infinite number of points to reduce wear and eliminate slippage. The clutch-coupling absorbs power surges when the motors are started.

The coupling portion of the over-running clutch is attached to the motor shaft and the clutch portion to the gear box. One motor can operate while the motor on the opposite side is shut down.



Pushing a button energizes a solenoid valve that vents hydraulic oil to a hydraulic motor. The motor rotates a dial which turns until desired speed is displayed. Hydraulic pistons in the dial selector mechanism move the gear shifters, providing . . .

Push button speed

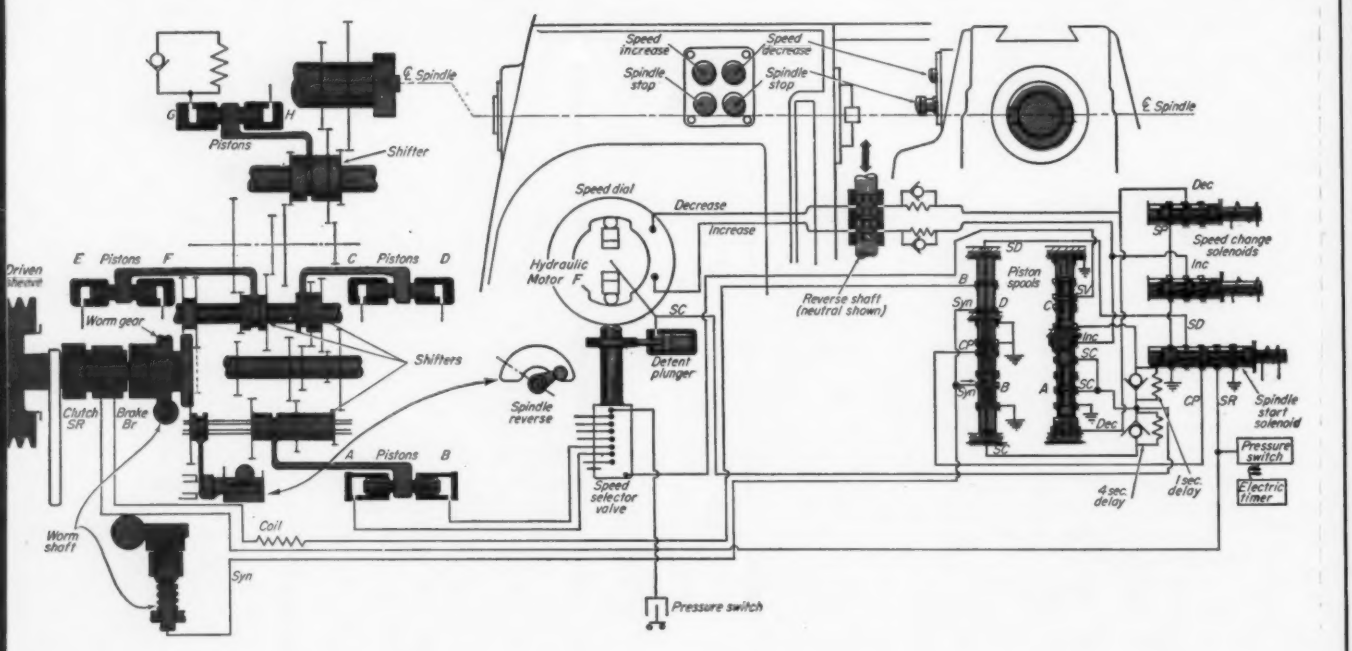
By **ALBERT DALL**, *Chief Engineer, Machine Tool*

SPINDLE SPEEDS ARE CONTROLLED by push buttons in a new dial-type milling machine. There are four buttons each on the front and rear control panels to permit operation at either point. "Start", "stop", "increase" and "decrease" buttons perform all the speed functions.

When the stop button is pushed, the clutch part of a clutch-brake disengages the spindle drive and the brake engages, preventing the gear train from rotating. Simultaneously, a synchromesh unit engages which, through a worm and wormwheel, causes the gear train to rotate very slowly. This assures clash-free meshing during the shifting operation. *The actual shifting is done when either the increase or decrease buttons are depressed. This energizes a solenoid valve that lets oil under pressure rotate a hydraulic motor.* A dial, attached directly to the motor shaft and clearly visible to the operator, rotates with the motor to indicate the desired rpm.

As the motor rotates, portings at intervals around the valve at the rear of the motor, line up with mating hydraulic lines. These operate hydraulic pistons located at both sides of the gear shifter arms. When the dial rotates to a new speed and the push button is released, the hydraulic pistons move the shifter arms to obtain the speed.

The synchromesh unit prevents gear clash by allow-



selection

Division, Cincinnati Milling Machine Company

ing the gears to mesh while being shifted. The spindle's start button will not respond until the gears have been shifted. This button releases the brake, disengages the synchromesh unit and engages the clutch permitting rotation at selected speed.

Push button control encourages the operator to use all the capabilities of the machine. He will select the correct speed for the job at hand rather than use the speed selected for the previous runs.

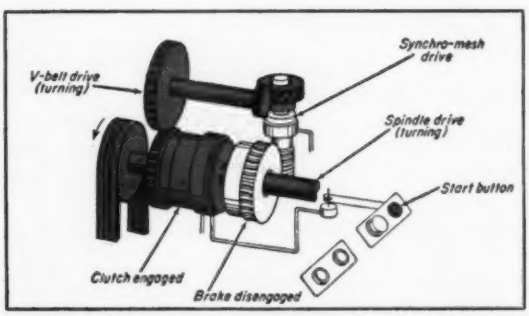
The spindle is powered by a separate motor that runs continuously. The motor drives the spindle shaft with a V-belt drive that turns continuously until engaged or disengaged by the clutch or brake.

Neither the increase or decrease button will respond to a change signal until after the spindle stop button is depressed. When speed changes are being made, pressure created in the hydraulic line causes interlock valves to travel to top position and raise detent plungers on the selector valve. This balances pressure in the hydraulic motor.

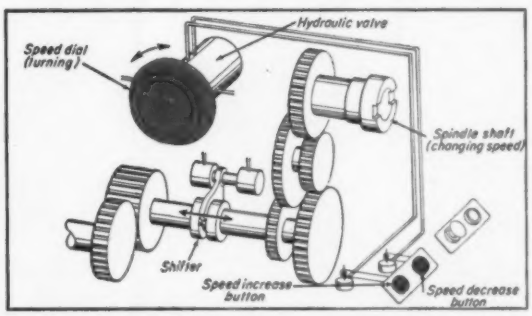
Referring to the schematic, displacement of valve spool B vents control pressure to the reservoir. This prevents the spindle start button from responding until after the change gears have shifted and spool B returns to low position. The upper position of spool B also applies pressure to line SYN making all gears rotate so that they can mesh while shifting. The displacement of spool C cuts off pressure in line SV to delay the shift until the spindle has had time to stop.

Nomenclature For Schematic

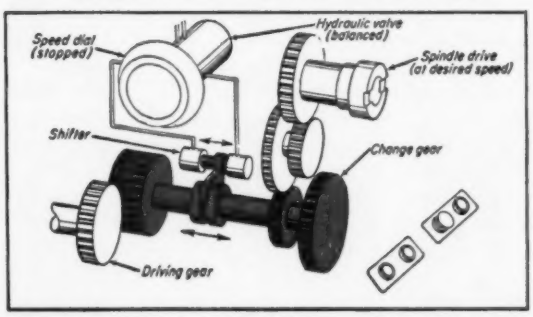
A,B,C,D	**= spools of interlock valves**
Br	**= brake**
CP	**= control pressure**
DEC	**= speed decrease**
F	**= hydraulic motor**
INC	**= speed increase**
SC	**= speed change**
SD	**= spindle start to valve D**
SR	**= spindle run**
SV	**= selector valve**
SYN	**= synchromesh**



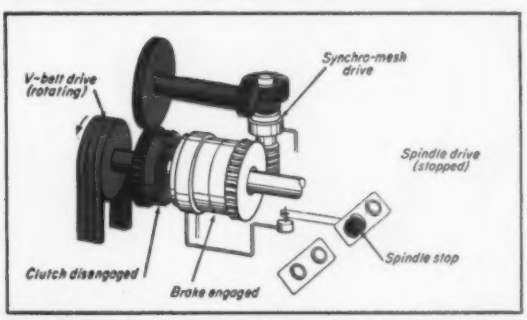
START BUTTON engages clutches and disengages brake applying instant power to spindle shaft.



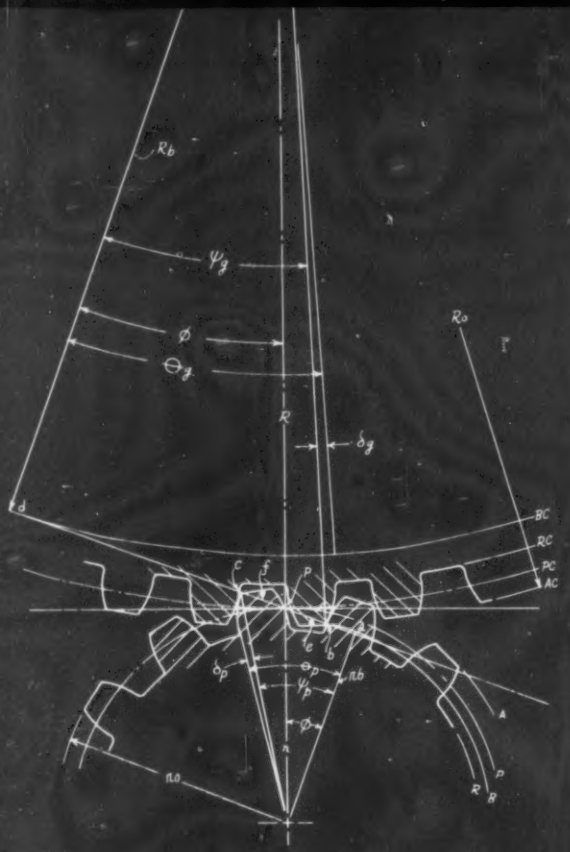
SPEED INCREASE and decrease buttons activate hydraulic drive which rotates speed dial.



WHEN BUTTON IS RELEASED at desired speed, drive unit rotates gears and shifts them.



SPINDLE STOP BUTTON disengages clutch and engages brake.



Eliminate the guess-

By E. K. GATCOMBE and R. W. PROWELL *Professors of*

Did you know that involute spur gears are the most used of all gear types? They maintain constant angular velocity ratios between shafts. Follow these computations to help you specify and use involute gears.

AN INVOLUTE IS DEFINED as the path traced by the end of a string as it is unwound from a circle. The circle is called the base circle. Involute spur gears transmit power only between parallel shafts. Their teeth are subject to a combination of rolling and sliding forces.

The fundamental law of gearing states that to maintain constant angular velocity, two mating teeth must have a common line that is normal to the point of contact and must intersect at a fixed point with a line drawn between the gear centers.

Let's determine all we can about a set of involute spur gears in a sample problem. Let's transmit 50 hp through a pinion at 1150 rpm with a gear set ratio of approximately 4 to 1. The center distance is approximately 7½ in. The gears will run continuously, 24 hrs a day.

Solution

Since the gears are for continuous duty, let us select a 20° stub involute. The straight tooth spur should be satisfactory for this relatively low speed. If the gears are cut with a reasonable degree of accuracy, the set should not be noisy.

To avoid interference and serious undercutting of the teeth, a pinion with more than 14 stub teeth is most satisfactory. Let us select an 18 tooth pinion.

Since we have a 4 to 1 gear ratio, the number of teeth would be 4 times 18 or 72. With this design the same pair of teeth engage every 4 revolutions of the pinion. The gear ratio does not need to be exact and if we make the gear with 73 teeth, the pinion will rotate 73 times and the gear 18 times before the same pair of teeth engage again. This extra tooth, called a hunting tooth, distributes the wear more evenly among the teeth. The actual gear ratio now becomes

$$M_g = \frac{N_g}{N_p} = 4.056$$

With a gear ratio of 4 and a center distance of 7.5 in.,

NOMENCLATURE

C = center distance, in.
D = pitch diameter of gear, in.
R = pitch radius of gear, in.
D_o = outside diameter of gear, in.
R_o = outside radius of gear, in.
R_b = radius of base circle of gear, in.
d = pitch diameter of pinion, in.
r = pitch radius of pinion, in.
d_o = outside diameter of pinion, in.
r_o = outside radius of pinion, in.
r_b = radius of base circle of pinion, in.
F = face width, in.
ϕ = (phi) pressure angle
θ = (theta) angle
Y = tooth form factor
ρ = radius of curvature of tooth profile on pinion, in.
ρ_g = radius of curvature of tooth profile on gear, in.
E_g = modulus of elasticity of material of gear, psi.
E_p = modulus of elasticity of material of pinion, psi.
e = maximum tooth error, in.
S = allowable bending stress, psi.
S_{es} = surface endurance limit, psi.
q_c = maximum contact pressure between teeth, psi.
b = half width of surface in contact, in.
Z_s = depth of maximum shear stress below surface, in.
T_{max} = maximum shear stress at depth Z_s, psi.
V = velocity of sliding, ft./sec.
ω_g = angular velocity of gear, rad-sec.
ω_p = angular velocity of pinion, rad-sec.

work in involute spur gears

Mechanical Engineering, U. S. Naval Postgraduate School

we have

$r + R = 7.5$ in; $\frac{r}{R} = \frac{1}{4}$

$\therefore r = 1.5$ in and $R = 6$ in

Rather than maintain an exact 7.5 in center distance, let us keep the pinion radius at 1.5 in and increase the gear radius. By so doing, we may be able to use a standard pinion. This pinion size is consistent with the required bore size.

To determine gear radius, first calculate diametral pitch P

$$P = \frac{N_p}{d} = \frac{18}{3} = 6$$

$$D = \frac{N_g}{P} = \frac{73}{6} = 12.167 \text{ in}$$

$R = 6.083$ in

$C = R + r = 6.083 + 1.5 = 7.583$ in

Next calculate and tabulate some of the tooth dimensions.

Addendum, $a = \frac{.8}{P} = 0.133$ in

Minimum Dedendum, $b = \frac{1}{P} = 0.167$ in

Working Depth, $h_k = \frac{1.6}{P} = 0.267$ in

Minimum total Depth, $h_t = \frac{1.8}{P} = 0.300$ in

Minimum clearance, $c = \frac{0.2}{P} = 0.033$ in

Circular pitch, $p = \frac{\pi}{P} = 0.542$ in

Base pitch, $P_b = p \cos \varphi = 0.492$ in

Basic tooth thickness on pitch line, $t = \frac{1.57}{P} = 0.262$ in

Outside diameter of pinion, $d_o = \frac{N_p + 1.6}{P} = 3.267$ in

Outside radius of pinion, $r_o = 1.633$ in

Outside diameter of gear, $D_o = \frac{N_g + 1.6}{P} = 12.433$ in

Outside radius of gear, $R_o = 6.217$ in

Next, calculate the pitch line velocity of the gears and the transmitted load on the gear teeth

$$v = \frac{\pi dn}{12} = 903 \text{ fpm.}$$

$$W_t = \frac{33000H}{v} = 1830 \text{ lb.}$$

A face width F of 3 in is selected consistent with the diametral pitch as well as the allowable force per inch of face width. Buckingham's equation for calculating the dynamic load $W_d = W_t + \frac{0.05v(FC + W_t)}{0.05v + (FC + W_t)^{1/2}}$

Let us specify class-two industrial gears which will have an expected tooth error of 0.001 in for a diametral pitch of 6.

$$C = \frac{k E_p E_g}{E_p + E_g} = 1720$$

$W_d = 4260$ lbs

Load normal to teeth $= W_n = \frac{W_d}{\cos\varphi} = 4530$ lbs.

Set $W_n = W_w$ and solve for the surface endurance limit.

$$W_n = W_w = dF \frac{S_{es}^2 \sin\varphi}{1.4} \left(\frac{2N_g}{N_p + N_g} \right) \left(\frac{2}{E} \right)$$

$$S_{es} = \sqrt{\frac{W_n \times 1.4 \times E\,(N_p + N_g)}{dF \sin\varphi \times 4 \times N_g}} = 139{,}000 \text{ psi}$$

This value of S_{es} calls for a steel pinion of 400 BHN mating with a steel gear of 350 BHN. The flexural endurance limit for this steel is 92,000 psi. To check the tooth strength, substitute this value into the Lewis Equation and solve it for W_s

$$W_s = \frac{SFY}{P} = 1740 \text{ lbs}$$

$W_s = W_n (1 + MS)$

$$M_s = \frac{W_s}{W_n} - 1 = 2.84$$

This margin of safety is more than adequate for even the heaviest shock loads.

Further Considerations

The velocity of sliding at the first point of contact

$$V = \frac{R_o \sin\psi_g - R \sin\varphi}{12} (\omega_g + \omega_p)$$

Continued on the next page

$$\cos \psi_g = \frac{R \cos \varphi}{R_o} = 0.918$$

$\psi_g = 23.1°$

$\sin \psi_g = 0.392$

$V = 4.7$ ft/sec

The velocity of sliding at the last point of contact

$$V = \frac{r_o \sin \psi_p - r \sin \varphi}{12} (\omega_g + \omega_p)$$

$$\cos \psi_p = \frac{r \cos \varphi}{r_o} = 0.86$$

$\psi_p = 30.05°$

$\sin \psi_o = 0.501$

$V = 3.8$ ft/sec

Since these values are quite low, no real danger of rapid wear is anticipated. The sliding velocity at the pitch point is zero and at other locations has intermediate values.

Two pairs of teeth will be transmitting the load near the beginning and ending of the contact. But, near the pitch point, only one pair will be carrying the load. The average number of teeth in contact is

$$N_{av} = \frac{bc}{P_b} = \frac{bP + Pc}{P_b}$$

$bP = R_o \sin \psi_g - R \sin \varphi = 0.36$ in

$Pc = r_o \sin \psi_p - r \sin \varphi = 0.307$ in

Average number of teeth in contact = 1.36

This figure is within the suggested value of 1.25 to 1.40.

Now calculate the Hertzian contact pressure for five different points along the line of action.

First point of contact.

$\rho_p = (R + r) \sin \varphi - R_o \sin \psi_g = 0.12$ in

$\rho_g = R_o \sin \psi_g = 2.47$ in

$$q_o = 0.418 \sqrt{\frac{WE (\rho_p + \rho_g)}{F \rho_g \rho_p}} = 122{,}000 \text{ psi}$$

First point of contact of a single pair of teeth.

$\rho_p = r_o \sin \psi_p - p_b = 0.330$ in

$\rho_g = (R + r) \sin \varphi - \rho_p = 2.260$ in

$q_o = 108{,}000$ psi

Pitch point

$\rho_p = r \sin \varphi = 0.513$ in

$\rho_g = R \sin \varphi = 2.08$ in

$q_o = 94{,}400$ psi

Last point of contact of a single pair of teeth

$\rho_g = R_o \sin \psi_g - p_b = 1.953$ in

$\rho_p = (R + r) \sin \varphi - \rho_g = 0.640$ in

$q_o = 83{,}600$ psi

Last point of contact

$\rho_g = (R + r) \sin \varphi - r_o \sin \psi_p = 1.77$ in

$\rho_p = r \sin \psi_p = 0.82$ in

$q_o = 54{,}800$ psi

Contact pressures decrease as the point of contact moves along the line of action. Also the maximum contact pressure is less than the surface endurance limit of 139,000 psi which was built into this set of gears. In view of this we would not expect these gears to be in any great danger of pitting.

From these computed values of the contact stresses, the stress of 122,000 psi found for the first phase of contact is highest when two pairs of teeth are making contact. Many test on gears have shown that pits usually begin to form between the base and pitch circles on the pinion. No doubt this relatively high stress condition is an important contributing factor. However, pits may form between the base and pitch circles. This would correspond to a region where the contact stress is relatively low. This represents a region where lubrication is less favorable than for other areas of contact.

The half width of the surface of contact at the first point of contact which is the worst condition is

$$b = 1.52 \sqrt{\frac{W \rho_p \rho_g}{F E (\rho_p + \rho_g)}} = 1.68 \times 10^{-3} \text{ in}$$

The depth of the maximum shear stress below tooth surface

$Z_1 = 0.78b = 1.31 \times 10^{-3}$ in

The magnitude of the maximum shear stress

$T_{max} = 0.304 q_o = 37{,}100$ psi

This value of $T_{max} = 37{,}100$ psi is well below the endurance limit in shear for the material for either the pinion or gear. The flexural endurance limit was 92,000 psi. The endurance limit in shear for this material would be approximately 50,000 psi. This gives a reasonable factor of safety. The favorable effects of lubrication have been neglected in computing these contact stresses.

The PVT relationship for the first point of contact

$P = 122{,}000$ psi; $V = 4.7$ ft/sec; $T = bP$

$PVT = 206{,}000$

which is well below 1,500,000.

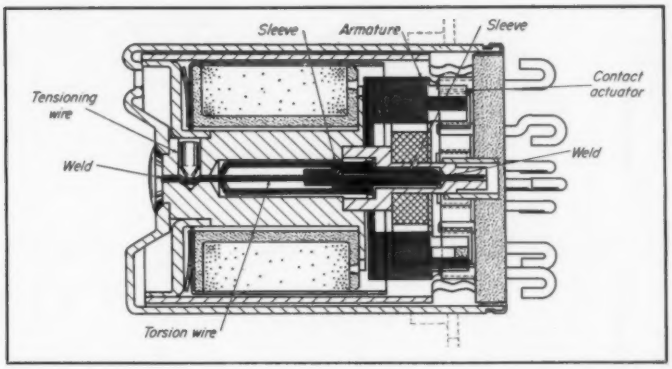
Summary

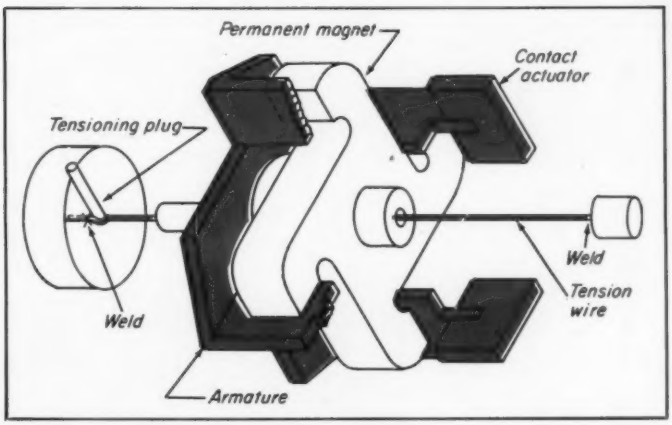
In summarizing, this set is operated at a reasonably slow speed. The amount of power to be transmitted is quite high. The contact stresses are of the magnitude usually found in industrial gears, 122,000 psi at the first point of contact of the pinion and gear. This is reasonable for industrial gears, but it may run much higher in other types, for example, in aircraft gears it may run well over 200,000 psi and in rocket motor gears it may go as high as 300,000 psi.

The maximum velocity of sliding of 4.7ft/sec is relatively slow and this velocity of sliding when considered in conjunction with a reasonably low contact stress and a small value of T in the PVT formula, results in a PVT product well below that value required for scoring.

IDEAS

for designers and plant engineers





FOUR-POLE RELAY. Cross section, upper left, shows position of magnets and tensioning wire inside shell. Perspective, lower left, illustrates their relationship. Entire relay is seen at right as it appears completely assembled before installation.

Tensed wire eliminates shaft and bearings

WEAR AND FRICTION ARE MINIMIZED by using a tightly-stretched wire as the armature support in a new 4-pole, 200 C relay. The wire prevents high-temperature problems that sometimes occur with pivot bearings, such as distortion and binding.

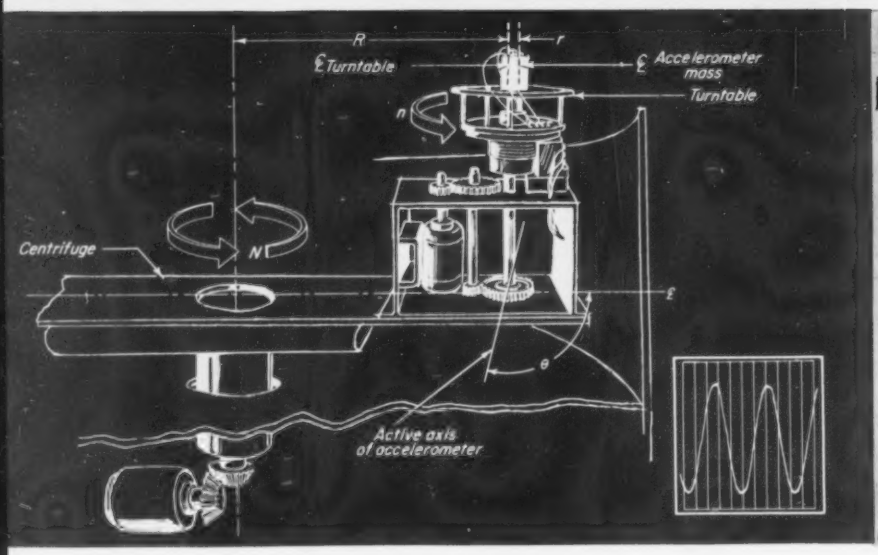
Wire sleeves through which the wire runs are attached to the armature. Ordinarily, the sleeves do not touch the wire. There is a small clearance between the wire and sleeve. But when vibration or shock move the armature, the sleeves contact the wire and holds the armature in position. Even when the sleeves touch the wire, there is practically no friction because the friction torque acting on the wire is at a radius of only about 0.010 in.

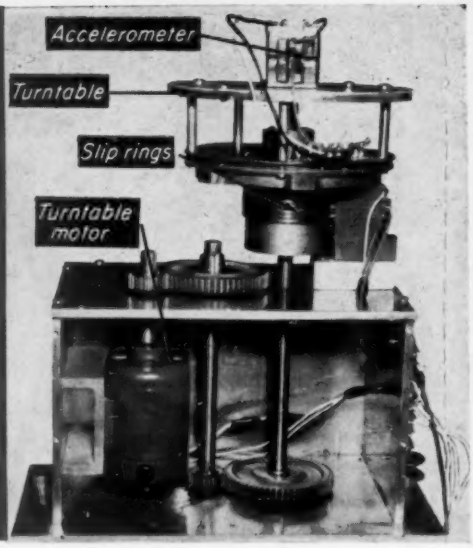
Rotation of the armature is less than 6 deg. This stresses the supporting wire to less than 3% of its tensile strength. With the wire welded to a support at one end, a small weight is applied to the free end to obtain the desired tension. A small wedge-plug is then inserted to maintain the tension. Tungsten wire is used because it has high tensile strength and good high-temperature resistance. Normal tensile strength of 600,000 psi decreases to 300,000 psi at 600 C. Torsional elastic modulus decreases only 5% at 700 C.

An electromagnet rotates the armature to operate four sets of contacts. Each set has one normally-open and one normally-closed contact. A permanent magnet moves the armature to off position when voltage is removed.

The relay is a development of Union Switch & Signal Div., Westinghouse Air Brake Co., Pittsburgh.

6 more pages of ideas





Eccentric motors produce sine wave forces

TWO MOTORS, one powering a centrifuge arm on which the other is mounted, produce forces of zero to 10 *g* from zero to '2 cps on a point on a turntable driven by the motor on the arm.

The centrifuge rotation produces a constant outward force on the point. But the platform rotation causes the point to see sinusoidal *g* loads.

This kind of force is needed to determine the response of accelerometers under these conditions. The centrifuge produces a constant linear acceleration, but rotation of the turntable changes the orientation of the sensitive axis of the accelerometer with respect to this acceleration and causes sinusoidal *g* loading of the accelerometer mass.

Force produced has two components. That produced by the centrifuge can be found from

$$g_1 = 2.84 \times 10^{-5} N^2 (R + r \sin\theta) \sin\theta \quad \ldots\ldots\ldots\ldots\ldots (1)$$

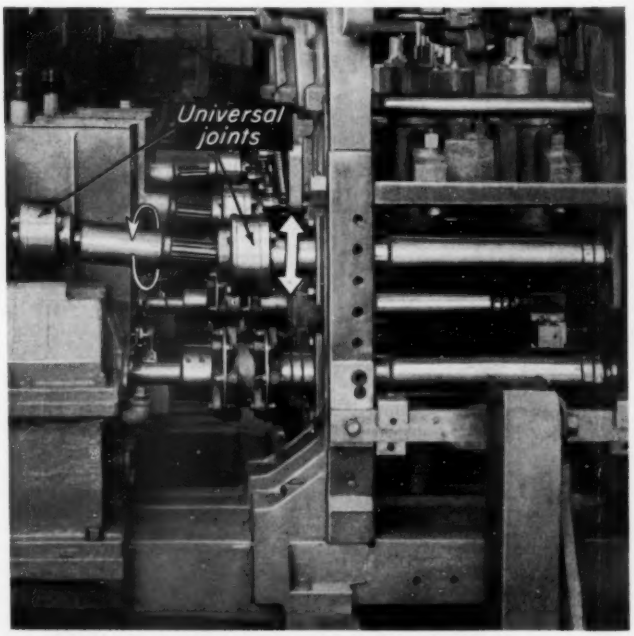
where N = centrifuge rpm; R = length from center of centrifuge to center of turntable, in.; r = distance from turntable center to accelerometer mass when

Constant-speed universals stabilize roll velocity

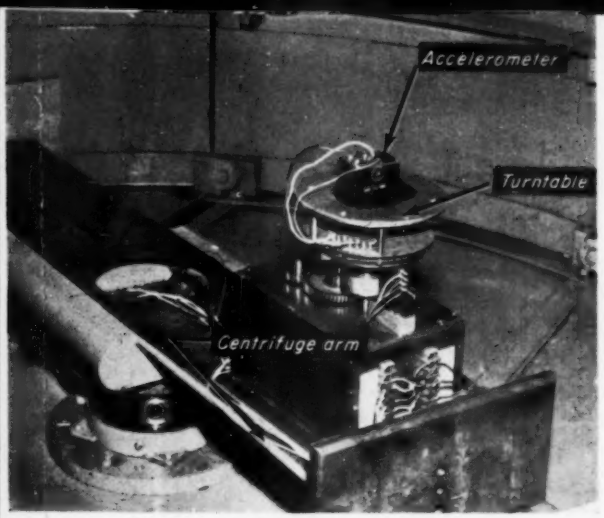
CONSTANT-SPEED universal joints keep speed constant and permit top roll adjustment in an aluminum rolling mill. Because conventional universal joints produce speed variations unless the shafts joined by the universal shaft are parallel, these constant-velocity universal joints are used. Speed variations would cause imperfections in the form being rolled.

Roll stands are powered by 2-speed motors that deliver 1.5 hp at low speed and 3.0 hp at high speed. A differential gear box drives upper and lower rolls because roll sizes may be different. The universal joints will operate at a maximum angle of 15 deg. A splined slip joint in the universal shaft lets shaft length change as roll spacing is changed.

Sutton Engineering Co., Bellefonte, Pa., makes the mill.



UNIVERSAL JOINTS can be seen above in position allowing rolling mill roll-adjustment.



continued

SINE WAVE FORCES are produced by eccentric motors. Schematic seen at left shows force-lines in graph. Photos illustrate how the unit is assembled when ready for use.

$\theta = 90$ deg, and θ = angle between centrifuge force axis and accelerometer mass-motion axis. Since this force displaces the accelerometer mass from the center of the turntable, the second force is that caused by rotation of the mass about the turntable center, and is

$$g_2 = 2.84 \times 10^{-5}\ n^2 r \sin\theta \quad \text{.....(2)}$$

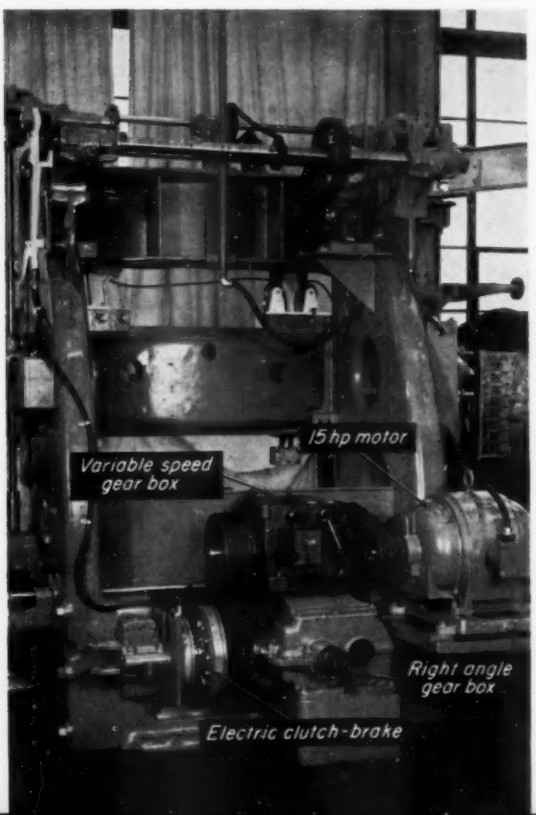
Summing equations 1 and 2, and combining terms gives total force.

$$g_T = 2.84 \times 10^{-5} \sin\theta\ (N^2R + N^2 r \sin\theta + n^2 r) \quad \text{.....(3)}$$

This shows that for constant values of N, n, R, and r, force varies sinusoidally.

Equation 3 shows that the n^2r component, which is undesirable, increases rapidly as n increases. Ideally r should be zero so this term and the $N^2r \sin\theta$ term drop out. Since r is not zero, correction must be made for displacement of the accelerometer mass due to the applied force. This also limits usefulness of the setup to a maximum of 10 cps with potentiometer accelerometers. With strain gage or variable-inductance accelerometers, higher frequencies can be used since total mass movement is very small.

The Instrument Group at Bell Aircraft Corp, Buffalo, developed this system.



Clutch-brake fits short space needs

TIGHT SPACE REQUIREMENTS on a 60-in. vertical boring mill were met by a combination electrical clutch-brake. A V-belt sheave integrated with the clutch also helps to keep size down, fitting between bearings 13 in. apart. Control buttons for clutch and brake are mounted on a pendant station. Start, stop, job, and complete disconnect controls are used. Electrical controls and the clutch-brake unit saved about 1500 lb of levers, clutches and other drive components.

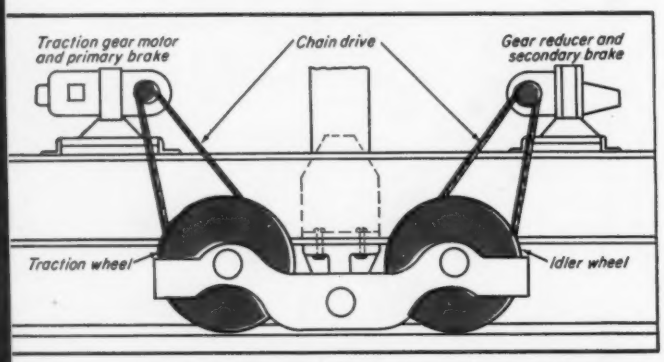
Other parts of the drive system are: 15-hp electric motor, change-speed gear box, multiple V-belts, and sheaves. Clutch input is to the main driveshaft which drives the mill through the original gearing.

The drive was redesigned by Botwinik Brothers Inc., New Haven, Conn.

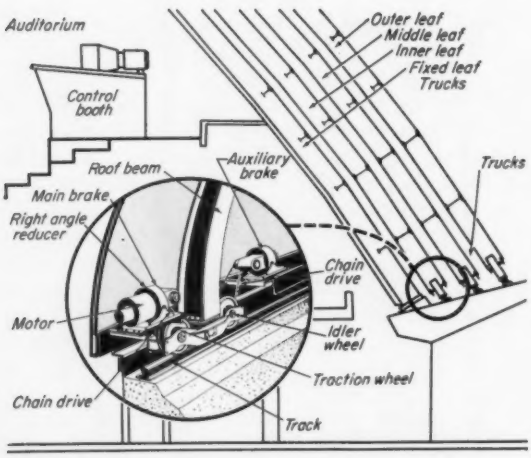
CLOSED roof has all leaves in position.

OPEN roof has leaves folded.

Gear motors synchronize different wheel speeds



DRIVEN AND IDLER wheels carry supporting trucks.



CHAIN DRIVE and truck in detail, in insert.

THIRTY GEARMOTORS open or close three pairs of auditorium roof-leaves, moving each pair at different speeds for various distances—completing each operation in exactly two and one-half minutes.

Pushing the "Open" button causes the motors (a-c 420 volt, 3 phase, 60 cycle) to turn at 1530 rpm. Ten are 25 hp, have a reduction ratio of 11.39, and move the two outermost leaves a distance of 135 degrees.

The second ten are 20 hp, have reduction ratio of 17.09, and move the intermediate pair a distance of 90 degrees.

Third group moves the low pair 45 degrees and has 10 hp with ratio of 38.44.

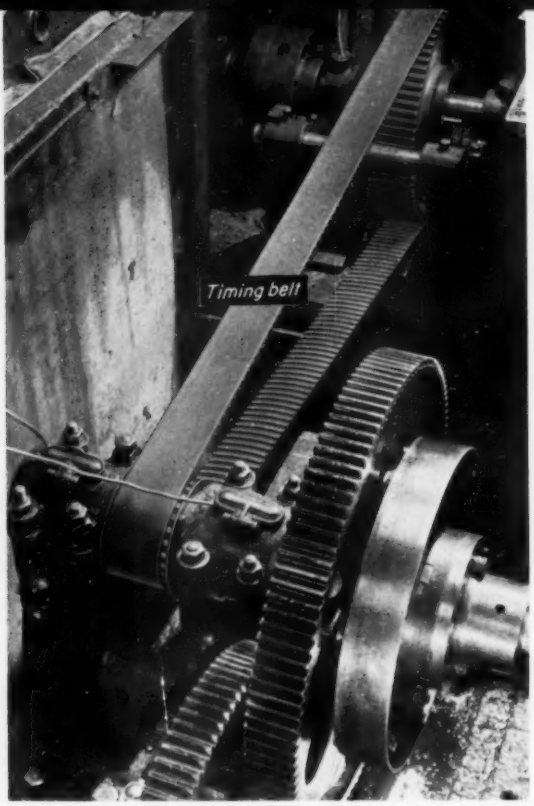
When the operator pushes the "Close" button, the motors reverse, moving the leaves together until they touch.

An electrical brake is flange-mounted on each motor. The idler wheel on each truck is chain-connected to the auxiliary electro-magnet brake. Thus, while only one wheel is driven, both are braked equally.

The trucks support the leaves at their lower surfaces. Limit switches start braking action after each pair of leaves is propelled its specific number of degrees. One wheel of each truck is motor-driven, one an idler. Design details are:

70,000 lb moving wheel load
120,000 lb static wheel load
7500 lb lateral thrust per wheel
Two spherical roller bearings per wheel
232.8 fpm (speed of top leaves)
150.0 fpm (speed, intermediate leaves)
75.0 fpm (speed, lower leaves)

The trucks are built for the Public Auditorium Authority of Pittsburgh and Allegheny County by Heyl and Patterson, Inc., Pittsburgh, Pa.



Timing belt stops vibration

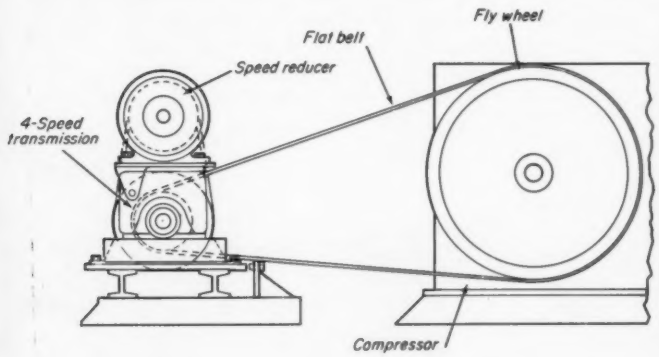
A TIMING BELT having teeth equipped with nylon-duck facing and backed with continuous helically-wound steel cables, solved several drive-problems of a newspaper plant's main press.

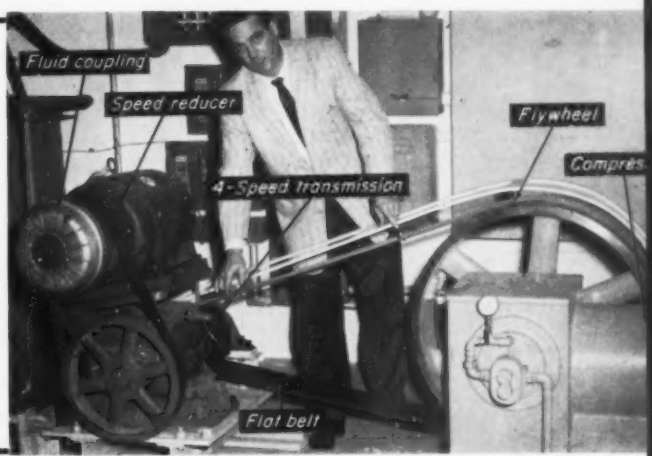
Advantages gained are: vibration is reduced; press now runs at top speed; more accurate two-color printing; chain-lubrication eliminated; quieter operation; smoother starting; cleaner motor pit.

The chain drive, once used, had the disadvantage of causing costly breakdowns and shaking the press. It was noisy and ran at less than full-speed.

The belt and pulleys have 7/8 in. pitch, 4 in. width. Belt has pitch-length of 175 in. Driver pulley has pitch diameter of 7.2415, the driven pulley has pitch diameter of 23.396. The unit is powered by a 30-hp motor, using a 10-hp starting motor.

The installation was made at Spokesman Publishing Co., Inc., New Brunswick, New Jersey.





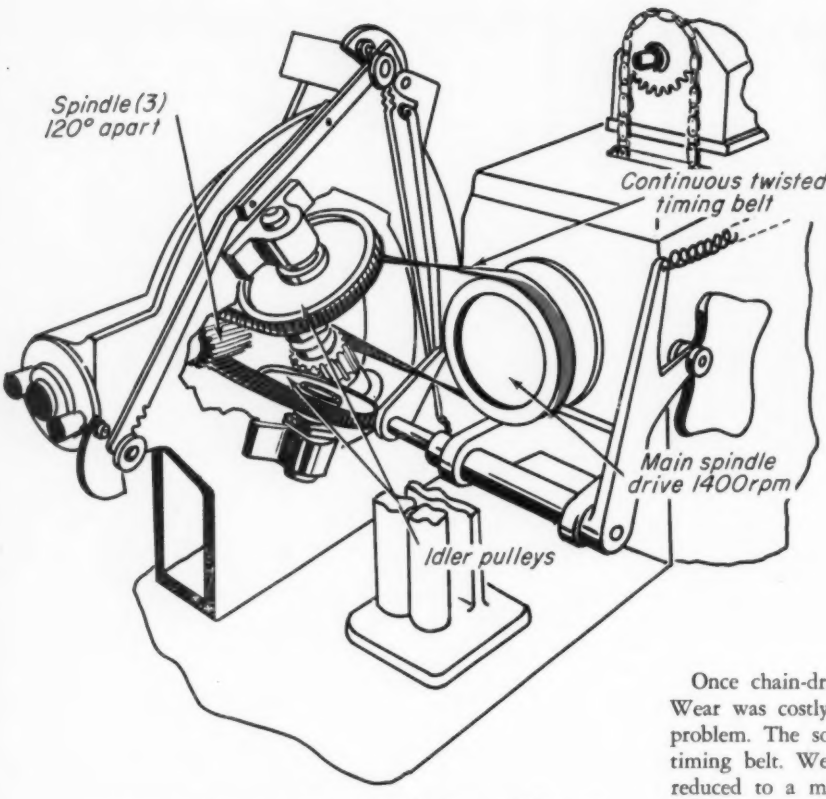
Flat belt improves low speed efficiency

FLAT BELTING provides better traction and higher efficiency between driver and driven pulleys on a liquid oxygen compressor. Driven by a 20 hp, 1750 rpm, a-c, synchronous motor, through a four-speed transmission, the belt turns a 33-in by 10-in flywheel as load varies from high to low. The wheel operates the compressor.

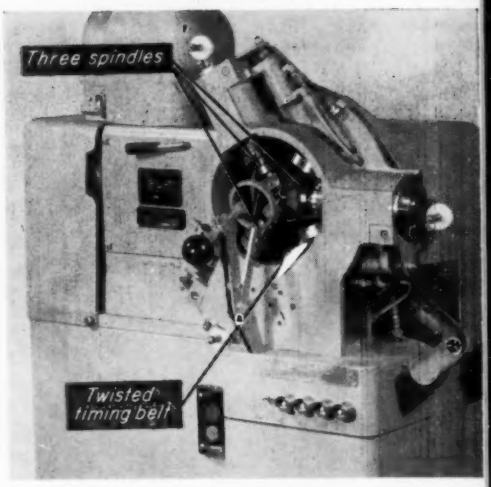
Formerly a V-belt drive was used but it was not practicable because of the difficulty of maintaining belt-tightness. Problem leading to selection of the flat belt was the need for variable speed control of compressor operation. This is necessary because pumping liquid gas through the vaporizer, where it is converted to a vapor, must be controlled.

Pressure in the compressor and load on the belt varies according to the amount of gas being pumped in to cylinders, the number of cylinders being filled, and the temperature of the cylinders.

Too much vaporized gas, forced through a system too fast, causes back-pressure, raising temperature of the system. This application is located at Burdett Oxygen Co., Columbus, Ohio.



TWISTING A TIMING BELT around three spindles synchronizes the speed of three armature coils so that they have an equal number of strands when wound on a winding machine.



Twisted timing belt

Once chain-drives and right-angle gears were used. Wear was costly and noise was a serious production-problem. The solution was installation of the rubber timing belt. Weight was reduced, noise abated, wear reduced to a minimum of six months. Another, unlooked-for advantage was halving the driving spindle rpm because of the weight reduction.

Other advantages of the timing belt: 85% reduction in maintenance cost; 75% reduction in lubrication

V-link belt stops long span stretch

THREE-STRAND V-LINK BELTS eliminate oven-time losses by resisting stretching and twisting while powering a cookie-cutting machine.

Open-end V-belts were used first. Stretching, twisting, slippages resulted because of the 53-in reach between motor and driven pulley.

Shut-downs for repairs resulted. Worse yet—oven-production time was lost—a loss which cannot be regained.

V-link belting is hard, does not stretch or twist. Strong raw-hide washers gave much extra strength. Punched to fit under the head of a long rivet, they add resistance to shock loads by distributing pressure over the entire link.

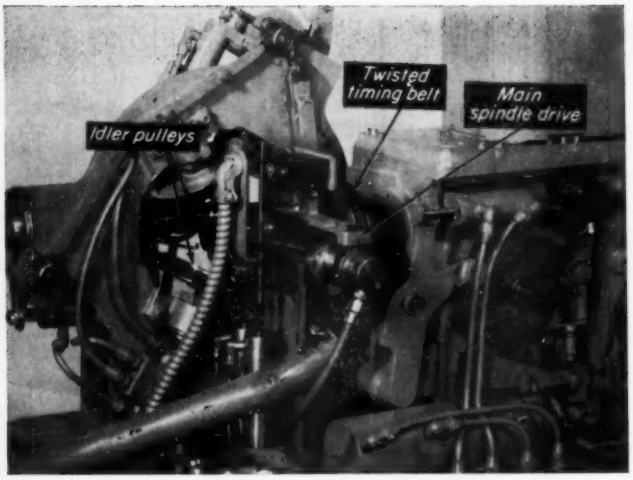
A 3 hp, induction motor drives a 4-in diameter, three-groove sheave. The C-type belt reaches 53-in from drive-sheave's center to the 11-in diameter driven-sheave.

The driven sheave drives a 5:1 variable speed block belt transmission. Speed is reduced further by a three-gear train driving a second 5:1 variable speed block belt transmission.

This is an especially illustrative example of an application of the long-reach, V-link belt. Because of the distances involved and the many cross-shafts, it was impossible to place the 3 hp induction motor any closer to the three-groove sheave. Flat belting would be unadaptable. The stretching and slippage, met with when former belting was used, rule out any adaptation of the earlier method.

This present application of V-link belting can be used in connection with other types of long reach, belt-drive operations where equipment placed between motor and sheave interferes. It offers the plant engineer a suggested solution for other power transmission problems.

The belt is used on ovens at the Burry Biscuit Corp., Elizabeth, New Jersey.



THREE SPINDLES, each set at an angle of 120° from the other two, are synchronized by a twisted timing belt. Belt's path through the machine is seen in drawing at far left. Machine, center, has its part in position. Photo at right gives closeup of belt, pulleys, and drives.

synchronizes three spindles

cost; much easier assembly; maintenance was facilitated.

Biggest was reduction of downtime. Four hours were once needed to take machine apart for repairs. Belts are slipped on in 10 minutes.

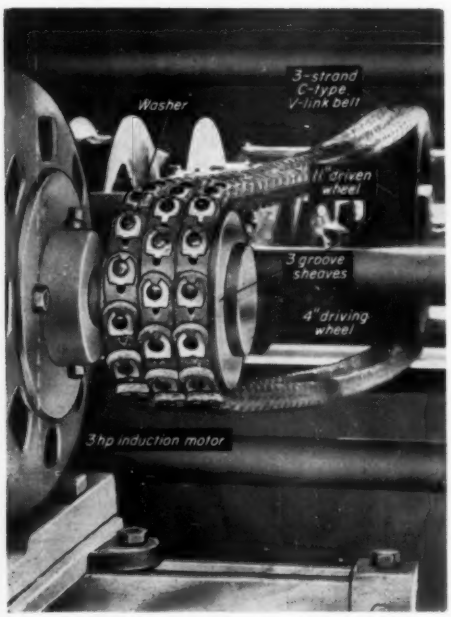
The unit is powered by a ½ hp, a-c motor turning at 1725 rpm. The unit is manufactured so as to be used with either 110 or 220 volt motors. Both are single phase but three-phase is available in the 220 or 440 volt.

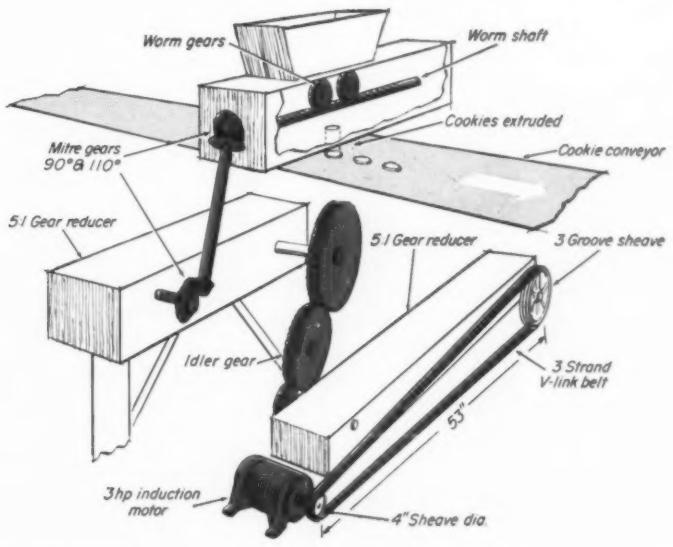
The drive turns at 1400 rpm. Driven pulleys turn at 2800 rpm, are 2-in in diameter, and the idler pulleys are 4-in in diameter.

At first glance, this application of the twisted timing belt has the appearance of unworkability. But an examination of the unit and its winding power-transmission medium reveals a new, different use of a familiar item.

The entire mechanism is an example of the results of an analytical approach to a problem of noise, high maintenance cost, expensive breakdowns, and other factors. These include over-all weight of the moving shafts and gears, using more power than is now needed.

This winder is manufactured by the Globe Tool and Engineering Co., Dayton, Ohio.





V-LINK BELT transmits power from 3 hp induction motor, along 3-strand belt, to operate the cookie-cutting gearing above.

LETTERS

Address letters to:
the Editor
Power Transmission Design
812 Huron Road, Cleveland 15, Ohio

Where's page 86?

Editor:

In the January 1959 issue of POWER TRANSMISSION DESIGN, I became interested in your article "Unbalanced Thrust Loads" by Norman M. Wickstrand, on page 66. This article is incomplete in my issue. It does not continue on page 86 as indicated.

Can you furnish me with the complete article? Or, let me know where I may obtain one?

ARTHUR F. ANDERSON
Corporate Engineering
Gar Wood Industries, Inc.
Wayne, Michigan.

• *Sir:*

Many of our readers also took the trouble to write about this omission in our January issue. We extend our apologies to the author, Mr. Wickstrand, of the Torrington Co.

Here is the rest of the article, just as it should have appeared when continued from page 78.

Editor

Page 86

UNBALANCED THRUST LOADS

Continued from page 78

tive center of the load is fairly near the centerline of the pump and bearing. Using needle thrust bearing, with a pitch radius of 1.290 in., the equation can be written

$$a/r = .263/1.290 = .204$$

Using equation (14) for F_1

$$F_1 = 1 + 2\,a/r = 1 + 2 \times .204 = 1.41$$

Effective load can be found from equation (11)

$$P_e = F_1\,P = 1.41 \times 2270 = 3200 \text{ lbs.}$$

If inlet pressure had been neglected, or if it had been zero, effective load on the bearing could be obtained from table 1

$$a/R = 0.7490$$
$$R = 2.125/2 = 1.062 \text{ in.}$$
$$a = 0.7490 \times 1.062 = 0.796 \text{ in.}$$
$$r = 1.290 \text{ in.}$$
$$a/r = 0.796/1.290 = 0.617$$

Then interpolating in table 2

$$F_1 = 2.311$$

Using equation (15)

$$P_e = \frac{\pi(.750)^2}{4}\,3\,\frac{950}{\cos 13^\circ}\,2.311 = 2990 \text{ lbs.}$$

Assuming the pitch diameter of the pump and bearing are essentially equal, calculation is more simple. Equation (17) and proper factor from table 3

$$P_e = 6.82\,\frac{950}{\cos 13^\circ}.750^2 = 3740 \text{ lbs.}$$

Nomenclature

- a = distance from reference axis to centroid—in.
- d = diameter of pistons—in.
- F = factor which depends on number of pistons and eccentricity of load
- F_1 = factor which depends upon eccentricity of load only
- h = maximum height of ungula (maximum load intensity)
- N = total number of pistons
- N_1 = number of pistons loaded
- p = discharge pressure of pump—psi
- p_1 = inlet pressure of pump—psi
- P = total actual load—lbs.
- P_e = effective load—lbs.
- r = pitch radius--in.
- R = pitch radius of pistons—in.
- y = height of ungula at any point
- α = variable central angle
- β = angle of load
- 2λ = limit of load
- θ = central angle between adjacent pistons
- δ = angle between axis of pump body and axis of upper thrust bearing

Editor:

In reading your January, 1959, issue, we were interested in an article on adjustable sheaves by Mr. Albert M. Rockwood.

However, the issue did not continue this article past page 27. Would you please send us tearsheets covering this complete article for our information. Also, we would like to know who is making this transmission.

D. J. WALLACE
H. A. Davis Power Equipment Co.
Dallas, Texas

• *Sir:*

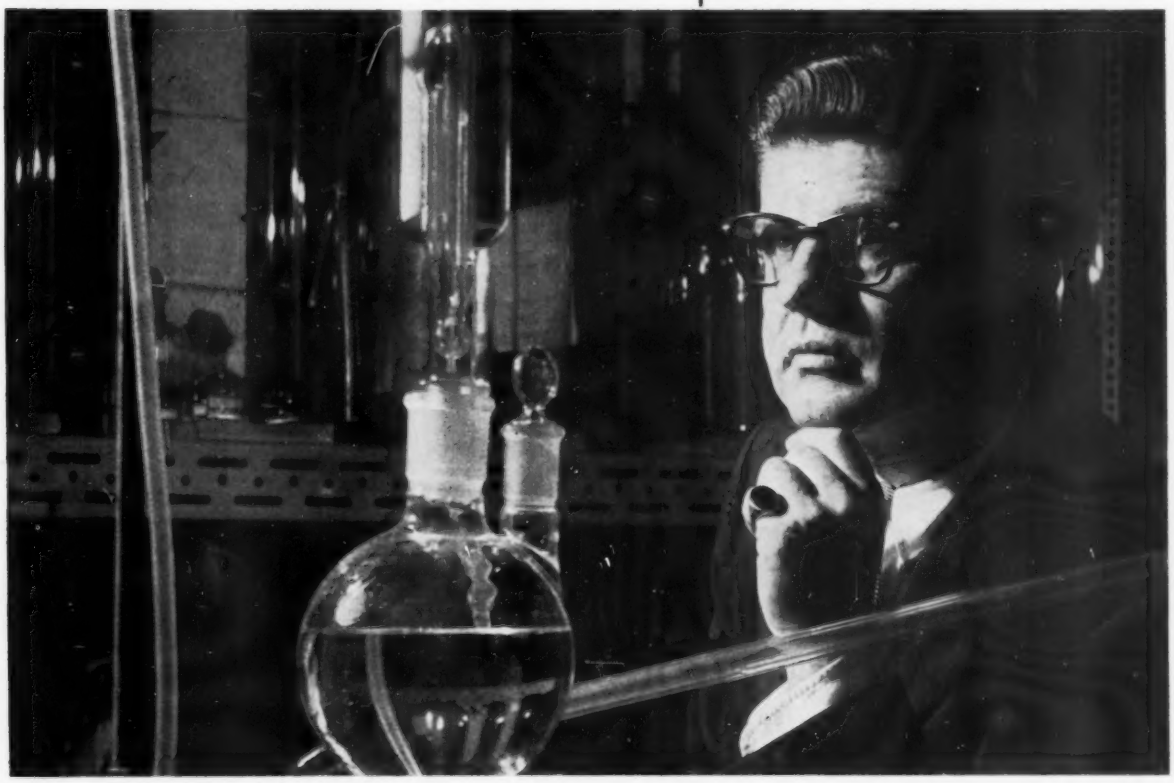
Only one line was omitted from this article, "Adjustable-pitch sheaves", *page 26, January 1959 issue,* POWER TRANSMISSION DESIGN. *It read: Patents are held by the Battelle Development Corp., Columbus, Ohio.*

Editor

Continued on page 60

He runs the only distillery in the business press

His name is David Kinsler. He distills *writing*. Every line in all eleven IPC magazines goes through the still he heads. When it comes out, it's *Distilled Writing*.



Why Distilled Writing? Because Mr. Reader is a busy man. If he read everything that crossed his desk, he'd never get his job done. So he reads magazines that are easy to read.

He reads POWER TRANSMISSION DESIGN. The editorial is distilled—extra words, but none of the facts, are squeezed out. What's left makes clear, concise, bright reading. Distilled writing makes articles shorter, leaves room for more articles to attract more readers . . . . helps readers get more facts quicker.

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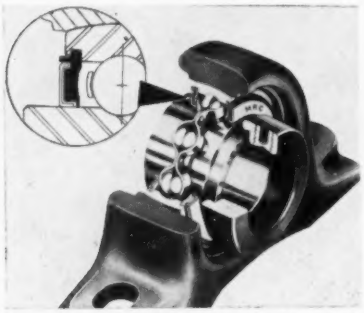
Pillow blocks

Permanently-lubricated pillow blocks using ball bearings pre-lubricated at assembly, eliminate need for any post-assembly lubrication. Seals holding lubricants in bearings combine advan-



tages of: rotating flinger; labyrinth seal; and positive contact synthetic rubber seal. The new pillow block permits adjustment of the bearing to shaft misalignment while keeping full contact seating of the bearing in the housing. Available in standard shaft sizes up to 1-15/16 in.

Marlin Rockwell Corp., Jamestown, New York

Circle No. 250 on the Reader Service Card

Auxiliary switch

Can be individually adjusted in steps of 15 degrees in the field without disassembling. Developed for circuit breakers and switchgear applications, it allows shorter, narrower stackings for space savings. It is now available for all secondary control voltages of 24, 48, 125, 250 volts d-c; and 220, 440 volts a-c.

Allis-Chalmers Mfg. Co., Milwaukee, Wisconsin.

Circle No. 260 on the Reader Service Card

Miniature clutches

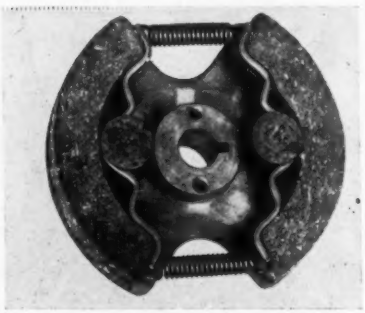
Basic assembly consists of four bearing steel rollers, and a cam block, both hardened and precision ground; four springs, and two side retainers. Complete clutches containing roller-assembly; housings; shaft; and in some models, precision ball bearings, are available as stock units. Stock roller assembly sizes range from .1875 inch bore, .625 inch O.D.; to .625 inch bore, O.D. Stock units are available in needed sizes and ranges.

Miniclutch Co., Hamden, Conn.

Circle No. 251 on the Reader Service Card

Centrifugal clutch

Cam-type, it uses moderate camming or locking-in action that allows it to disengage at the same speed it engages, with or without load. It gives quick release when necessary and has been



adapted for use where gradual engagement is required. With motors, low-line voltage protection is provided. It can be adapted to various drives such as: pulley, sprocket, gear, or coupling. When the drive is mounted on the driven member, clutch may be used as a power take-off.

Fairbanks, Morse, and Co., Magneto Div., Beloit, Wisconsin.

Circle No. 270 on the Reader Service Card

Miniature shaft coupling

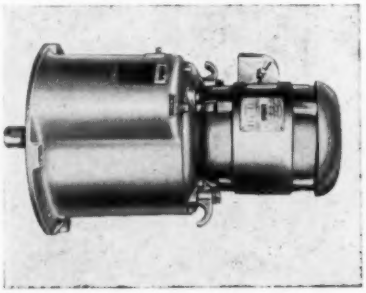
Weighing 1/30 oz, has 1/4 inch diameter, 1/4 inch length, and is made for 1/8 inch shaft. Helical design allows compactness and gives smooth bearing-loads, constant velocity, and absolute zero-backlash. Each is ground from a single piece, thus there are no pins, solder-joints, screws or rivets. Integral flanges, gears, and clutch components can be incorporated into spelings. They have many specialized uses which make them especially adaptable to various applications.

Helical Products Co., Hermosa Beach, Cal.

Circle No. 258 on the Reader Service Card

Gearmotors

Vertically mounted, they are supplied with double or triple reduction units in ratings from 1 hp to 75 hp, and with reductions from 350 rpm to 7-1/2 rpm. The units operate at nor-



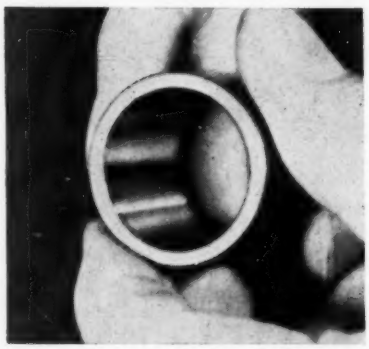
mal torque on 2 or 3 phase, 60 cycle, 208, 220/440 or 550 volts. Special voltages and frequencies as well as mechanical modifications are available.

The Lima Electric Motor Co., Inc., Lima, Ohio.

Circle No. 249 on the Reader Service Card

Sleeve bearing

Made up mostly of iron powers, physical properties are: Porosity, % oil by volume, 18-23; Ultimate Tensile Strength (psi), 10,000; Compressive Yield Point (psi), 9500; Rockwell Hardness, R_h-35; Specific Gravity, 6.1-6.5. Suggested uses include bearing



and bushing applications on home appliances, fractional horsepower motors, power tools, light machinery, and instruments. Bearings are sold under trade name Iron Oilite 212.

Amplex Div., Chrysler Corp., Detroit, Mich.

Circle No. 259 on the Reader Service Card

Spherical bearings

Outstanding among new developments, these spherical bearings have such added advantages as anti-friction qualities. Other features that must be pointed out are such highlights as the surfaces

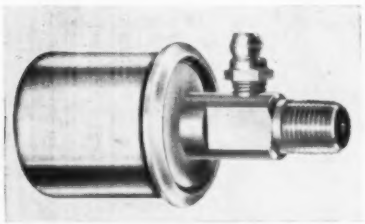
treated with Teflon. Another feature is the application of sintered material on the surfaces.

U. S. Bearing Corp., North Hollywood, Cal.

Circle No. 257 on the Reader Service Card

Bearing lubricator

Measuring 2-1/4 inches by 1-1/8 inches, automatic lubricators are installed, one to each bearing by the simple act of screwing automatic lubricators into place. Each has a lubricant reservoir, automatically injecting grease as needed. Mountable in cramped positions,



it uses standard fittings, will operate in any position.

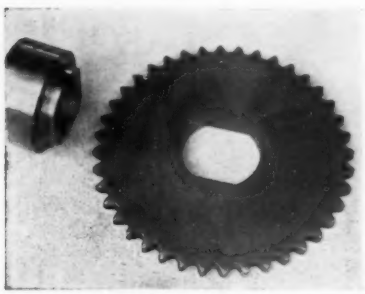
Lubrication time saving is 75%. Automatic signals tell when reservoir needs filling.

Anchor Chemical Co., Cleveland, Ohio.

Circle No. 256 on the Reader Service Card

Sprockets

Die cut sprockets can be furnished complete with hubs of any shape. They also can be furnished with a driving hole. They may be staked on the hub if desired. Also, they can be furnished with either a plain round hole known as type "A", or a plate sprocket. They are made from 1020 steel. Heat treating is offered at slight extra cost. The sprockets can be had in various sizes and tooth numbers.



Dayton Rogers Mfg. Co., Minneapolis, Minn.

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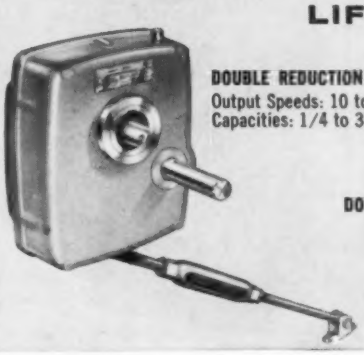
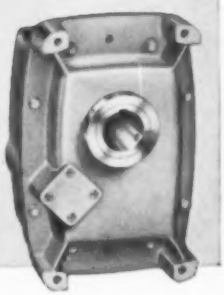
SINGLE REDUCTION
Output Speeds: 90 to 420 RPM
Capacities: 1/4 to 40 HP



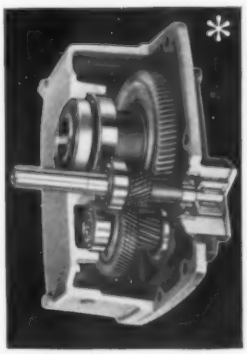
DOUBLE REDUCTION
Output Speeds: 10 to 160 RPM
Capacities: 1/4 to 30 HP



DOUBLE REDUCTION FLANGE MOUNT
Output Speeds: 10 to 135 RPM
Capacities: 1 to 7 HP

Foote Bros. Shaft Mounted Drives offer more efficient, more economical, power transmission. They incorporate exclusive Duti-Rated Lifetime Gearing — the high hardness, balanced design, premium quality gearing that combines greater load carrying capacity with long service life.

Used with standard V-Belts and Sheaves, Foote Bros. Shaft Mounted Drives will provide virtually any output speed you may require. Quick, easy installation on driven shafts with diameters from 15/16" to 3-7/16" saves time, labor . . . eliminates need for reducer mounting, couplings, and adjustable motor mount. Built-in Backstop to prevent reverse rotation, Automatic Overload Release Torque Arm, Variable Pulley, are available as optional equipment.



The complete Foote Bros. Shaft Mounted Drive line is made in accordance with all applicable AGMA Standards

Write for NEW SHAFT MOUNTED DRIVE CATALOG

100 YEARS SERVING INDUSTRY 1859-1959

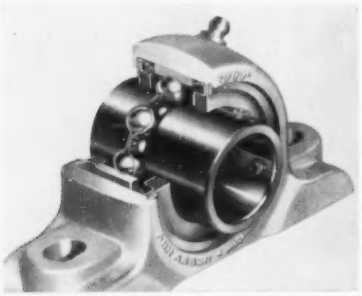
FOOTE BROS.

Better Power Transmission Through Better Gears

FOOTE BROS. GEAR AND MACHINE CORPORATION
4575 SOUTHWESTERN BOULEVARD • CHICAGO 9, ILLINOIS
For more information circle No. 18 on the Reader Service Card

Pillow block bearings

Made with an extended inner ring,



two set screws, and elongated bolt holes, these bearings can be locked to the shaft and are easily mounted. Units are available in shaft sizes ranging from half-inch to 1-3/16 in. Also in the new line are units with heavy cast iron housings for normal duty.

Hoover Ball and Bearing Co., Ann Arbor, Michigan.

Circle No. 252 on the Reader Service Card

Fractional variable-speed drive

New line of fractional variable-speed drives from 1/8 hp to 3/4 hp are all-electric and have 8 to 1 speed range. They are designed for driving small conveyors, grinders, printing presses, welding positioners, packaging machines, small agitators, and other applications. They feature a single half-

wave thyratron electronic tube as a main rectifier, operate on 50 or 60 cycle, 220 volt a-c power. Weight is 30 lb. Measurements: 9-1/2 inches, by 14 inches by 8-1/4.

Reliance Electric and Engineering Co., Cleveland, Ohio.

Circle No. 253 on the Reader Service Card

Variable-speed drive

In ratings from 3/4 hp to 7-1/2 hp, with a stationary field construction. Brushes, slip rings, and commutators are eliminated. Length is reduced 22% and both the a-c motor and eddy current clutch are built into a common housing. In the Adjusto-Spede drive an a-c motor drives a clutch drum at constant speed, while speed on the output member is adjusted by varying d-c excitation to the clutch-coil. Direct current is supplied by a small controller. The drive is designed to supply precise operating speeds for machine tools, process machinery, test equipment, conveyors, printing presses, and similar equipment.

The Louis-Allis Co., Milwaukee, Wisconsin.

Circle No. 266 on the Reader Service Card

Stepping motor

Two miniature, high-speed models are

available. Both have uni-directional and bi-directional units. The stepping switch is a 10-point circuit precision switch. Its contacts are plated for minimum size and maximum reliability. Bi-directional feature is possible because of two identical stepping motors which drive the same shaft—but in opposite directions. Output shaft rotates 36° for each input-impulse up to 40 steps per second. Both units can be had in standard voltages of 24, 48, and 90 vdc, added to this is 110 vac. Direct current models have maximum input power rating of 12 watts; alternating current units, 15 watts.

Data Instruments Div., Telecomputing Corp., North Hollywood, Cal.

Circle No. 254 on the Reader Service Card

Motor bases

Small-motor bases have 4-1/2 inch handwheel and screw adjustments with lock-nuts to limit movement and assure settings without slippage. Rigid guide rods give smooth sliding action. Model SB-2L permits 5 inch bed travel, SB-2S permits 4 inch travel. The first model, 7-1/2 inch by 17 inch, handles

1/2 hp, 3/4 hp, 1 hp motors. These bases allow wide transverse movement of fractional horsepower motors with variable-pitch sheaves.

Rampe Mfg. Co., Cleveland, Ohio.

Circle No. 269 on the Reader Service Card

Motor

Direct current 28 volt motor operating at 69 amperes, delivers 1.6 hp at 2000 rpm. Other features: It is explosion proof, having flame-quench rings per MIL-E-5272; mounting flanges with other flanges or shafts available; radio noise suppression filter per MIL-E-6181; and, capable of withstanding continuous starting at rated load, at 2 second intervals, under severe inrush

conditions. Actuator operation is at 260°F ambient temperature, at 60,000 feet. For higher ambient temperatures of 500°F and 200,000 feet, d-c motors are available for missile applications.

Hoover Electric Co., Columbus, Ohio.

Circle No. 261 on the Reader Service Card

Speed reducer

Provides infinitely variable speed from 0 to 400 rpm, constant torque 60 or 100 in-lbs from 1/2 or 3/5 hp motors respectively. Speed reducers feature re-

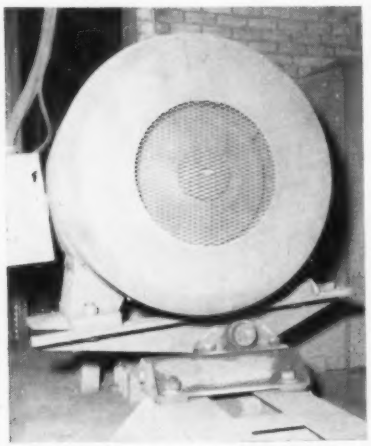
peating of speeds and instant speed changes, whether units are running or not. They will also serve as clutches. Output shafts are free wheeling in direction of drive and will drive in either direction. Basic dimensions are 6 inches by 8 inches by 12 inches. Speed reducers are available, with or without motors. The units are designed for use on powering feeds, positioners, conveyors, machine tools, and in blending and measuring. They will operate in any position.

REVCO, Inc., Minneapolis, Minn.

Circle No. 265 on the Reader Service Card

Motor base

Matching tension to load, maintaining belt alignment, eliminating slip, and



giving automatic take up—are characteristics found in the Tens-A-Matic Pivotal motor base. Maintenance on belts and bearings is reduced. Sizes are available for motors ranging from 1 hp to 200 hp.

Industrial Marketers, Ferndale, Michigan.

Circle No. 264 on the Reader Service Card

Contactor

Immersed in oil, the 400-amp contactor has 50,000-kva interrupting ratings for either 2.5 or 4.8 kv. It is designed to handle 2500 hp squirrel-cage motors at 4800 volts and unity power factor synchronous motors up to 3000 hp, 4800 volts. Medium voltage controllers equipped with the type 426 oil immersed contactor.

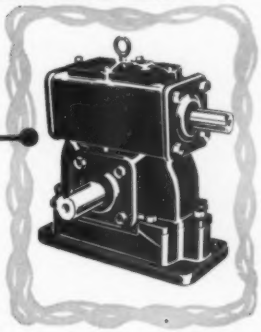
Allis-Chalmers Mfg. Co., Milwaukee, Wisconsin.

Circle No. 262 on the Reader Service Card

For more information circle No. 1 on the Reader Service Card

Abart

Precision-Built SPEED REDUCERS



- Fractional to 400 HP.
- Ratios to 10,000 to 1

A Size and Type for every Speed Reduction Need

Choose the speed reducer you need from Abart's complete line. Rugged, compact units are easy to install on new or old equipment—assure lower operating and maintenance costs. Worm, spur and combination gear—single or double reduction—with any desired shaft arrangement. Call your local Abart distributor today, or write direct for handy pocket-size speed reducer catalog.

Abart Custom-Cut GEARS

No Stocks • Any Material • Any Quantity

All Abart Gears are precision cut to *exact* specifications. Spur, spiral, bevel, helical, internal worm, rack, sprocket types—½ to 96 D.P., ¼" to 18" P.D. Request latest Abart Gear Bulletin. Send b/p and specs or sample for prompt estimate.

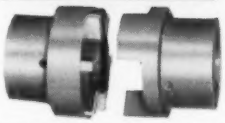
ABART GEAR & MACHINE CO.
4833 West 16th Street Chicago 50, Illinois

Gerbing

Cushion Type
FLEXIBLE COUPLINGS
No Lubrication Required

These flexible cushion insert type couplings compensate for misalignment of rotating shafts and effectively absorb shock and vibration. No metal to metal contact No lubrication, maintenance, or wearing of parts. Available in sizes from fractional to 700 HP. Inserts are available in various materials to meet all operating conditions and gov't requirements. Special bodies, bores, plating can be furnished.

Model GH*



Heavy-duty flexible coupling. Sizes from 35 to 700 HP (@ 1800 rpm). Has removable inserts held in place by steel collar to provide easy inspection or replacement of inserts. Available for shafts up to 5½" dia.

Model GHF*

Same as Model GH (above), except has flange as shown for use on gasoline motors and other drives where coupling must be bolted directly to flywheel, etc.

Model G

These exclusive concave jaw surfaces hold spider in position at all times —prevents insert from extruding under load.

Available for shafts ½" to 2¼" and HP ranges from fractional thru 30 at 1750 RPM. Can be installed quickly—only 3 parts; 2 metal hubs and a flexible spider. Unaffected by moisture, abrasives, dirt, etc.

**Models GH and GHF can be supplied with tapered bores to accommodate QD bushings.*

For detailed information request a copy of our Flexible Coupling Bulletin . . .

Gerbing manufactures a complete line of mechanical variable speed drives, fractional thru 25 HP.

Gerbing MFG. CORP.
11807 Milwaukee Ave.
NORTHBROOK, ILLINOIS
A Suburb of Chicago

For more information circle No. 23 on the Reader Service Card

Variable-speed sheaves

New line of wide-belt, smooth-faced sheaves which operate on fixed centers using 1-3/16 in V-belts. Combination of two sheaves and hand control provide low cost variable drive. Controllable sheave is mounted on the motor and the spring actuated sheave on the driven shaft. Available between 1 hp and 3 hp, speed ranges as high as 8 to 1. With 1750 rpm motor, output speeds of 400 to 3200 can be obtained.

Speed Selector Inc., Chagrin Falls, Ohio

Circle No. 228 on the Reader Service Card

Rotary actuator

New line of rotary actuators with modular construction having 40 sizes from 12 standard interchangeable parts. Called R-12, the units come in load ratings to 100 lb per in. and speeds from 0.5 through 50 rpm for aircraft use.

Airborne Accessories Corp., Hillside, N. J.

Circle No. 229 on the Reader Service Card

Hydraulic drive

Has 1½ hp reversible motor, speed range between 0 rpm and 750 rpm. Motor drives variable-displacement hydraulic pump with silent chain and sprocket. Handwheel controls speed selection. Reversing lever controls change of direction. Single or 3-phase, 60 cps, 220 v. Output shaft 5/8 in. Cost under $125.

Roberts Mfg. Co., Chicago. Ill.

Circle No. 226 on the Reader Service Card

Variable-speed sheave

MS series has new design that eliminates fretting corrosion, freezing and sticking. Bearing surfaces oiled by reservoir on each rotation. Continuous pumping action. No keys to obstruct flow. Series of Nylon-faced rubber keys transmit power from stationary to movable flange through sleeve cap. Motor base handwheel moves motor controlling speed.

T. B. Wood's Sons Co., Chambersburg, Pa.

Circle No. 227 on the Reader Service Card

Hold on there! Back up a bit: you just rushed by some pretty important information. Before you pass this copy of PTD on to the next reader check it over once more. Did you spot the New Products pages? See page 13. How about this month's selection of free data books and catalogs? They start on page 52. The handy Reader Service Card will bring you details on all of them. Just fill one out and drop it in the mail.

Gearmotor line

New line of integral gearmotors with both concentric and right angle shafts for integral horsepower motors through 100 hp. Also included is line of fractional right angle gearmotors. Available with open or totally-enclosed motors, double and triple reduction in standard ratios.

Allis Chalmers Mfg. Co., Milwaukee, Wisc.

Circle No. 330 on the Reader Service Card

Miniature permanent magnet direct current motors

Excluding shaft, BD is 2 9/16 in. long, rated 1/45 hp, 10,000 rpm, continuous. BL is 2 15/16 in. long, 1/30 hp. 10,000 rpm, continuous. Both are 1½-in.-diameter cylinders. Meet aircraft standards.

Globe Industries, Inc., Dayton Ohio.

Circle No. 205 on the Reader Service Card

Motion indicator

If you have a Roto-Guard on a shaft that goes too slowly or stops, the Roto-Guard will flip its switch and set off an alarm or whatever else you have it connected to. Measures 5 x 6 x 6 in., stainless steel ½-in. shaft. Adjusts to speeds of from 10 rpm to 150 rpm. Robustly built.

Bin-Dictator Co., Detroit, Michigan.

Circle No. 202 on the Reader Service Card

Clutch

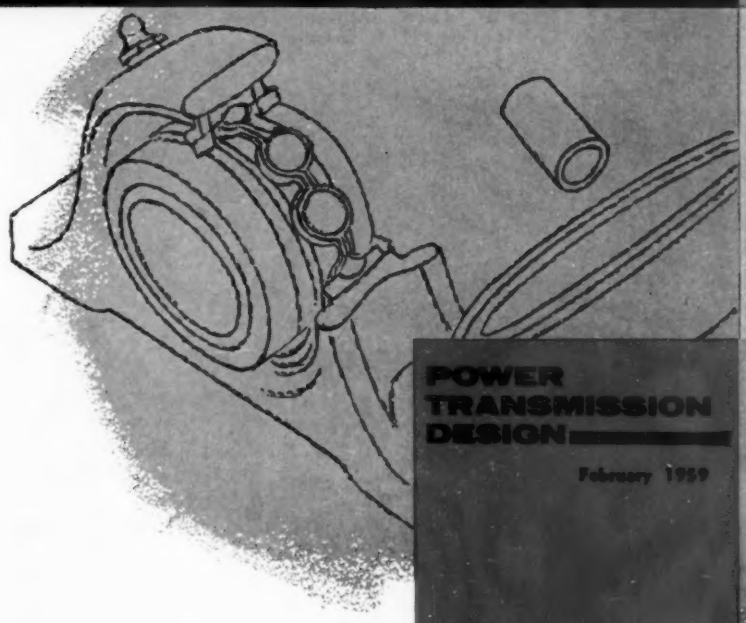
Unlike others, the Stationaire's air housing and piston rotate right with the clutch. A Stationaire can even be mounted right in the middle of a long shaft.

Conway Clutch Company, Cincinnati, Ohio.

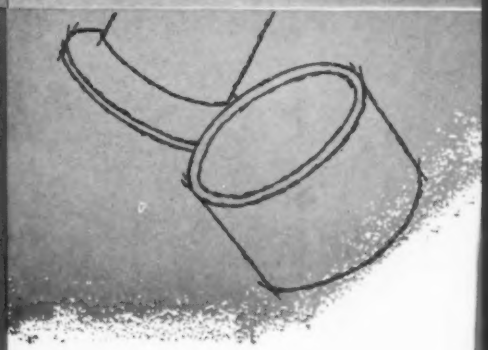
Circle No. 201 on the Reader Service Card

BEARINGS

DESIGN/APPLICATION



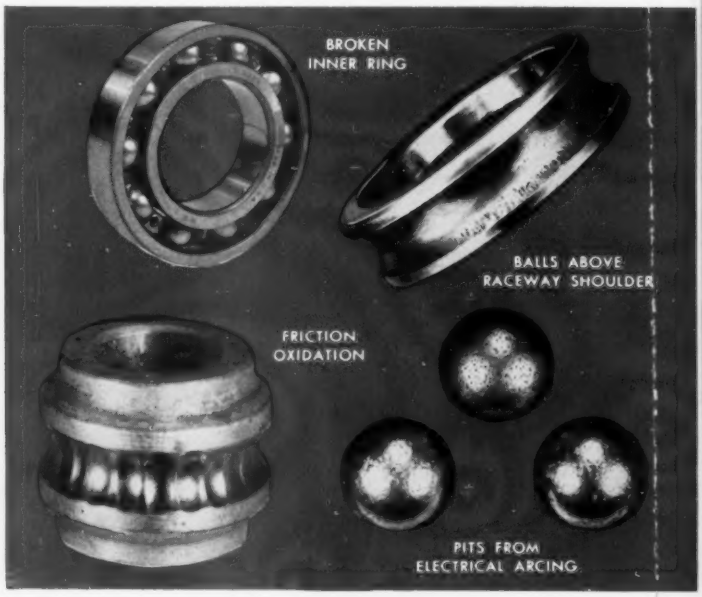
REGULAR MONTHLY SECTION OF IDEAS AND DEVELOPMENTS



POWER TRANSMISSION DESIGN

February 1959

Why anti-friction bearings wear



Improper installation and faulty lubrication ruin more bearings than any other cause. Here are some danger signals that may help you increase the life of your bearings.

CHOOSE THE RIGHT bearings for the right job. Install them properly and lubricate them in such a manner that dirt and other detrimental material will not get in. If you do this, you will get the maximum service out of not only bearings but your whole machines.

How do you know when bearings are worn out or need to be replaced?

Sometimes you don't, but it is usually obvious because of noise, excessive heat or malfunction of other rotating parts on the machine. When you dismantle, you will find any one of a series of problems. You may see pits in the raceways, on the balls or on the rolling surfaces. There may be small flaked areas, discoloration, roughness, broken retainers and a host of other signs of deterioration. Whatever the problem, there is a cause and usually it is difficult to analyze.

One cause is fatigue failure which occurs when metal surfaces flex until intracrystalline slip planes reach their ultimate ability to slide on one another. It is the same thing that happens when a piece of wire is bent until it breaks. Fatigue failure has also been called "crystallization" and can be prevented by keeping bearing loads below the ability of the metal parts to resist.

If flaked bearings are continued in operation, the size of the flaked areas will increase. The whole machine will become noisy, the shaft will become loose and other parts of the machine also will fail by resulting overload. *Conversely, if a gear or adjoining part fails, the overload may cause early fatigue failure of the bearing.*

Metal particles from an adjacent gear or mating part sometimes get in the lubricant and act as an abrasive and create a dull, gray appearance. This results in increased internal looseness of the bearing. The particles pressed between the rolling elements and the raceway may cause indentations and pits.

Occasionally, machines are assembled and stored in warehouses and are permitted to become damp while idle. Corrosion pits can roughen the races to make the bearings noisy and unserviceable. Under these conditions many lubricants oxidize and leave a thin layer of gum on the bearing surfaces making operation rough and irregular. *Some petroleum solvents or ether can be used to dissolve these gums to allow the bearings to continue in service.*

Bearings with flaked or roughened raceways impose extra work on the cages to force the rolling elements over the rough spots. This working causes flexing and decreases cage life. Breakage of one or more cage sections occurs. Continued use causes balls to skid in the raceways, frictional heat to build up and welding of the balls to the raceway surface.

out

By **HUDSON T. MORTON** *Chief Metallurgist, Industrial Tectonics, Inc.*

Faulty Installation

The most flagrant mistakes occur during mounting. If the outer ring is pressed when mounting, or if the inner ring or shaft are pressed when seating, the load might be applied through the balls or rollers. In either case, the thrust load may exceed the capacity of the bearing and "brinell" the shoulders of the raceway. To avoid such damage, the bearing should be supported by a small tube that fits the inner ring while the shaft is pressed into it. Be sure this tube or the blocks do not scrape threads or contact the outer ring.

If the distance between the end of the shaft and the bearing seat is short, hold the shaft in a vise and press the bearing on. When an arbor press is not available, tap with a hammer evenly around the tube. The tube should have a shelf above the bearing to prevent dirt or chips from falling into it. Tap lightly at first to make sure the bearing or ring goes on square and does not scrape or burr the bearing seat. *Be sure to tap the bearing to a firm seat on the shaft shoulder.*

For rotating shaft applications, the clearance between the bearing and housing should be sufficient to allow the bearing to slide into its housing seat. For rotating wheels and pulleys, tubes are generally used to push the outer ring tight against the bearing seat shoulder, in the same manner as for mounting bearings on shafts.

Improper tools may cock the inner ring on the shaft to cause expansion and breakage. If the tools slip they may damage seals or shields. Hammers or other tools used in driving bearings into position may crack or chip the shoulders rendering them unfit for use. When gears or other adjacent parts project, be careful that cages don't get damaged.

Duplex bearings and most angular contact bearings are stamped "THRUST HERE" on the face so that the high shoulder will carry the thrust load. If the thrust side is inverted, the balls may contact the low shoulder, become overloaded and show early fatigue flaking.

Likewise wheel bearings, tapered-roller and angular-contact ball bearings installed without removal of end play might fail at the low shoulder.

Misalignment

When bearings rotate with inner rings at an angle to the shaft axis, the balls wear a track which wobbles from one side of the raceway to the other. *Three common causes of this are: shoulder out of square, burr between shoulder and bearing, and fillet too large.* To overcome this, shoulders must be perpendicular to shaft axis without large fillets.

The same misalignment difficulties arise when the housing shoulders are not perpendicular to the shaft axis, or the shaft is not in line with the housing. Standard internal fit bearings (Code O) can permit a

Continued on page 47

Which type bearing has the least friction?
Lubrication plays the deciding role.
Here are the basic differences between . . .

Plain vs rolling-contact bearings

By **J. P. VIDOSIC**, *Professor, Georgia Institute of Technology*

TWO GENERAL TYPES of bearings may be said to exist: Plain (hydrodynamic and hydrostatic) and rolling-contact (which are sometimes called anti-friction).

Plain Hydrodynamic

The underlying relationship of this type bearing can be expressed in the pressure equation

$$\frac{1}{6\mu}\left[\frac{\partial}{\partial\chi}\left(h^3\frac{\partial\rho}{\partial\chi}\right)+\frac{\partial}{\partial z}\left(h^3\frac{\partial\rho}{\partial z}\right)\right] = u\frac{\partial h}{\partial\chi}+h\frac{\partial u}{\partial\chi}+2v.$$

Where
μ = viscosity of the lubricant
h = film thickness

The physical significance of terms on the right side is most important because it explains why and how hydrodynamic bearings operate.

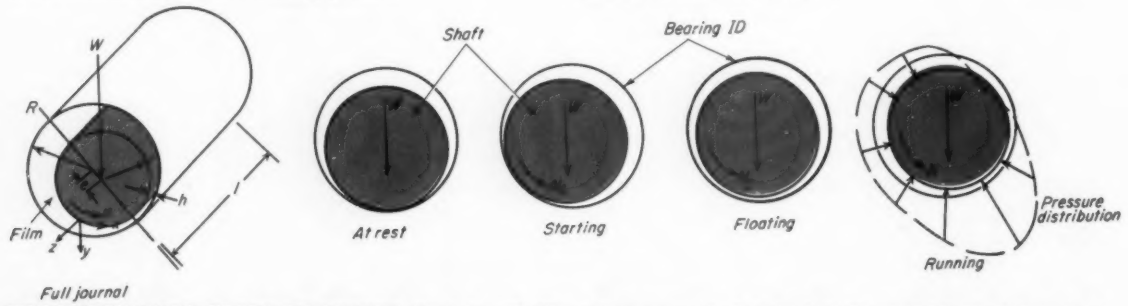
The first term depicts the *wedge* requirement. For pressure to be generated intrinsically within the bearing, capable of supporting a load, the lubricant must be wedge-shaped. Thickness h of the lubricant film must vary in the direction of flow which means the two surfaces moving relative to each other cannot be parallel. The angle, however, is very small, only a matter of a degree or two. The wedge forms as a result of the eccentricity of the running journal.

The *stretch* effect required by the second term is a progressive change in the relative velocity of the two surfaces. This, of course, does not exist in metal bearings. However, under certain conditions, dynamic loads provide this effect. The last term demands motion of the journal in the radial direction. This squeeze action occurs when the bearing is loaded dynamically. It is the last two terms which account for the situaiton that allows a bearing to support a larger dynamic than static load.

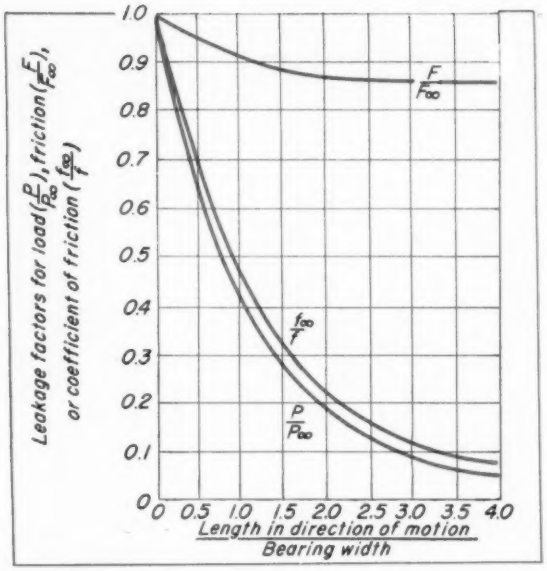
The exact solution of the three-dimensional equation is practically impossible. The addition of assumptions, especially that the bearing is of infinite length, reduces the equation to the simpler two-dimensional form

$$\frac{d}{d\chi}\left(h^3\frac{d\rho}{d\chi}\right) = 6\mu u\frac{dh}{d\chi}$$

Though only the wedge effect is left, the equation is still difficult to solve. However, it has been solved for full journal bearings as well as for some of the more-used partial journal bearings. Since an actual bearing is of finite rather than infinite length, correction for leakage factors must be applied. An ideal bearing is analyzed then corrected for end leakage. Typical of such leakage factors, usually defined as the ratio between equivalent bearings of finite and infinite length, are shown in the curve above.



HYDRODYNAMIC PLAIN BEARINGS float in oil film when properly lubricated. No metal-to-metal contact.



BEARING leakage factors to give minimum friction.

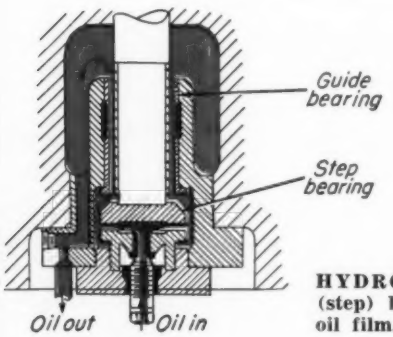
Characteristic	Hydrodynamic	Rolling
Axial space required	—	Less
Capacity	—	Better
Design	Easier	—
Dependability	—	Better
Diametral space required	Less	—
Initial cost	Less	—
Life as affected by wear	Same	Same
Lubrication requirements	—	Simpler
Maintenance	—	Simpler
Materials required	Simpler	—
Overload effects	Less	—
Running friction	Same	Same
Speed range	—	Wider
Starting friction	—	Less
Warning of failure	Greater	—

COMPARISON, plain vs rolling contact bearings.

The most important single property of lubricants in plain bearings is viscosity. Lubricants most commonly used are liquid, like petroleum base oils; plastic, like greases; solid, like graphite and molykotes; and gaseous, like air. Viscosity governs the effective friction which decides the power lost within the bearing. Although the coefficient of friction is very low in a properly-lubricated hydrodynamic bearing, heat is generated and the temperature of the lubricant rises. This rise in temperature must be limited, otherwise the lubricant may rapidily decompose and impair the operation of the bearing. Heat balance at a satisfactory lubricant temperature is therefore absolutely essential. When the bearing contains large surfaces or is well-ventilated, convection and radiation dissipate the heat generated. The oil, itself, carries the heat away. Other means of artificial cooling are also used on heavily-loaded bearings.

Plain Hydrostatic

When heavy loads are applied at low speed, conditions make hydrodynamic action unsuitable. This is because sufficient pressure to support the load cannot be developed as a result of the wedge, stretch or squeeze effects. Instead, if the bearing is to be kept free of damage and the power loss kept economical, external pressure must be provided. Such is the hydrostatic bearing. Shafts like those of large tufbines must have starting and stopping periods. To do this, the shaft must first be floated hydrostatically. This breaks metal-to-metal contact and substitutes a film of oil. Rotation can then proceed without any damage to either the shaft or the bearing. When full running speed is reached, pressure is then generated hydrodynamically and the external supply can be automatically cut off. The same steps in reverse take place on stopping.



HYDROSTATIC plain (step) bearing lies on an oil film.

The most interesting hydrostatic bearing is that supporting the Hale telescope at the Mount Palomar observatory in California. Close to one million pounds rotate with extreme smoothness at one revolution per day at a coefficient of friction of less than 0.000004.

Hydrostatic bearings are constructed in the form of pads like in the Hale telescope; and step bearings for vertical turbogenerators; lifts for large ball mills; journal bearings for heavy horizontal turbines; and oil cushions for reciprocating loads.

The basic phenomenon of hydrostatic action is like the flow of liquids in capillary tubes. The action is defined by the Hagen-Poiseuille law

$$Q = \frac{bh^3}{12\mu} \frac{dp}{dl} \text{ in}^3/\text{sec}$$

Where
b = width of slot
h = film thickness
μ = the viscosity
p = the pressure
l = the length of slot

The load which can be supported is found by adding the inlet pressure force to the force developed over the sill area where pressure is gradually dropping.

$$W = \rho_o A_o + \int_{po}^{o} \rho dA \text{ lb}$$

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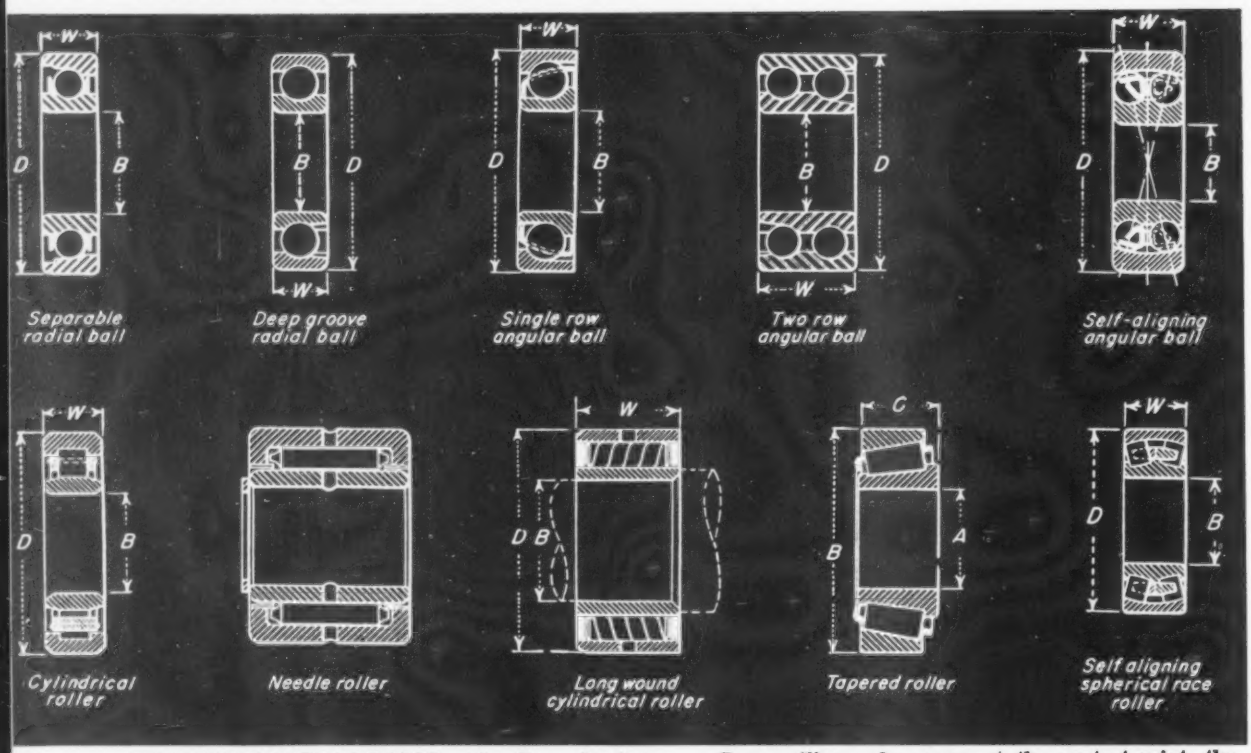


ROLLING-CONTACT BALLS and rollers also have sliding surfaces and depend on proper lubrication. Pure rolling only occurs at the contact-point, theoretically of zero area.

Temperature rise, which must be controlled, results from two causes. One is flow energy which is heat generated when a liquid is forced to flow under pressure. Contributing to this is energy generated as the film shears because of relative motion between bearing parts. Another factor is the increase of viscosity as pressure increases. Since pressures can be rather high (1000 psi or more) the viscosity factors used in the computations should be at the supply pressure.

Rolling-Contact Bearings

Rolling motion is known to occur with less friction than sliding motion. The standard rolling bearing, whether ball or roller, consists of four parts: the inner race or ring; the outer race or ring; the rolling element; and the cage or retainer. In the case of roller bearings, the roller can be cylindrical, tapered, hour-glass or barrel-shaped. Spiral roller and needle bearings are special types.

Proper operation of roller bearings depends greatly on choice and quality of materials from which they are made and precision with which they are manufactured. The actual design of bearing components is highly-theoretical and quite-involved mathematically. They are usually based on the Hertzian formulas which deal with the action of curved surfaces while in contact under load. Though his work did not apply specifically to rolling bearings, his deductions proved extremely valuable in bearing research.

Usually, the design engineer is concerned only with bearing selection. He chooses the best type and size of rolling-contact bearing for the application. To do this, he should understand a few facts.

Common ball bearings are made in three types: Radial bearings that support loads normal to the shaft; thrust bearings that support loads along the shaft; and angular-contact bearings that support combined loads, i.e., both radial and thrust.

Roller bearings are also made for radial and thrust loads and in the case of tapered-roller, combined loads can be supported. To increase capacity, roller bearings are mounted with two rows of rolling elements.

Special-type bearings are also made. Among these are the spherical-race and needle. Spherical race permits slight lateral rotation between inner and outer rings, allowing for small shaft misalignment. Where greater load capacity for least radial space is desired, needle bearings are widely used. Needle bearings can support radial loads only. They consist of a full complement of small-diameter rollers (needles) in an outer shell only or between an inner ring and outer shell.

Another special type are the miniature bearings made in the order of one millimeter in diameter. They are used in instruments, tiny motors and other small precise apparatus for delicate operation. Bearings used in inaccessible places for exacting requirements are sometimes provided with metal or felt shields between the outer and inner rings on one or both sides.

Continued on page 64

Lubricated-for-life bearings are highly dependable and widely used in electric motors, pillow blocks and applications where repeated greasing is difficult or undesirable. Here are a few pros and cons to help you decide when to use them

Life-lubricated vs conventional bearings

By W. A. WILLIAMS *Consulting Editor*

Why you should use life-lubricated bearings...

THE SECRET TO SUCCESSFUL long life of lubricated-for-life bearings is in the grease and seals. Few people know this better than the manufacturer. For this reason, bearing manufacturers can eliminate greasing and other maintenance factors as variables in lubricated-for-life bearings. They can apply the proper grease in the proper amount and give you a sealed-for-life bearing that will hold up in spite of you.

The best greases are usually used. These are usually non-oxidizing synthetic greases that have a life rating higher than all other types. They will last as long or longer than the bearing would have lasted under ordinary service.

The manufacturer will not take a chance using a cheaper grease that the user may deem satisfactory. He also uses the right amount and installs it under the most ideal conditions. There is virtually no possibility that contamination could occur. The user cannot introduce dirt through carelessness.

This type bearing is more simple and consequently less expensive than when external seals are used. This saving is usually realized at the beginning and is very attractive to users.

Too much grease can cause as much damage as too little. Improper type of grease can shorten life considerably. By using the best grease, if contamination is kept out, lubrication can be eliminated as a factor in short bearing life.

Goods seals can probably be called the most important single contribution to long sealed-for-life bearing life. There are three general types: *labyrinth, rubbing, labyrinth and rubbing combined.*

If conditions of severe contamination exist, such as in atmospheres of dust, salt water or steam, no bearing can be better than the capacity of the seal to keep out the contaminants. One manufacturer of pillow blocks recently presented data on salt water tests which showed that bearings with tight-fitting labyrinth and rubbing seals ran 24 days on-and-off under a spray of salt water; the best of four ordinary pillow block bearings, lubricated every day with water resistant grease, lasted only 8 days. There were some bearings that lasted only *ONE* day.

This and other tests prove an important point. *It is a false theory that, if you pump in new grease every day, you'll drive out the contaminants and keep the bearing properly lubricated.* This is not true. Even if it is, up to a point, think of the work and cost in grease and man-power keeping the bearing serviceable.

Some users contend that very wide bearings prolong life on the basis that a larger reservoir of grease provides extra lubrication service. This may be true.

The sealed-for-life bearing is built to be free of most of the human elements of maintenance. This is highly desirable and works out well in appliances and other consumer products where maintenance is impossible.

Why you should use conventional bearings...

YOU CAN SELECT from a wider range of sizes and types of conventional bearings than lubricated-for-life; and your needs are very different from someone else's. There are only a very few kinds of lubricated-for-life bearings now on the market. They are principally the deep-groove ball. All heavy-duty roller and sleeve bearings are conventional with lubrication fittings. Thus,

Continued on next page

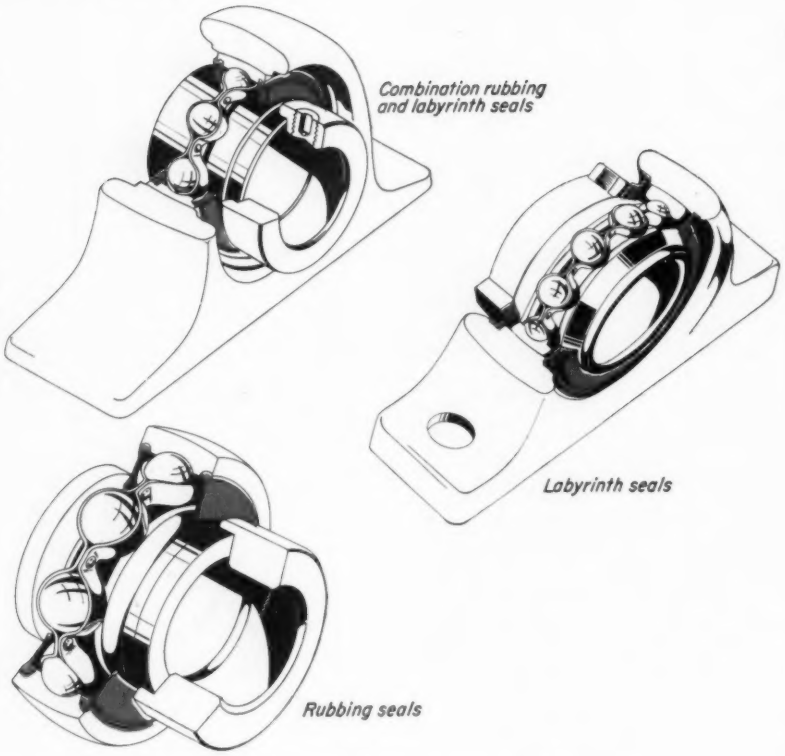
Three basic types of seals . . .

LASTING A MACHINE-LIFETIME, the bearing samples seen at right show high-quality grease seals.

Combination rubbing and labyrinth seals, see top, offer double protection, meeting conditions of high temperatures and other ambient variations and atmospheres.

Labyrinth seals, far right, impregnated and suspending lubricant in their miscroscopic filament-like orifices, give constant sealing action.

Rubbing seals, lower left, contacting raceway surfaces, give special advantages not found in others.



if you start mixing the two types, you may get into maintenance trouble. Maintenance of the two types is distinctly different. If you are in a place that has many bearings, such is in a large plant or in an airship, a conventional bearing might get the worst of it, if overlooked during a greasing. The man responsible might mistake one or more conventional bearings for the sealed-for-life kind.

High Temperature and Speed

Under conditions of high temperature (say 250 F and over) it is difficult to find a lubricant that will stand up. Repeated re-lubrication is mandatory to keeping the bearing in service.

At high speed, the life of the grease is also shorter, making frequent re-lubrication a necessity. As it stands today, permanent lubrication has not been proven practical.

For lightly-loaded bearings, at normal speeds, the life of the bearing is reduced by the life of the grease which is normally shorter than the bearing life. This restriction is undesirable and can be controlled by replacing the grease at regular intervals.

Conventional bearings need not necessarily have rubbing seals to keep out contaminants. Rubbing seals cause bearings to run hotter than labyrinth or ring seals. The higher the speed, the more pronounced the condition. This aggravates life factors since temperature is the worst foe of even the best grease.

There is a wide variation in the quality of seals available for bearings. Many users are not aware of this. A good seal for one application is not necessarily good for another. Sometimes you may not be aware of the difference. With conventional bearings your choice is greater and you can match the bearing to the application much more easily than with sealed-for-life bearings.

All manufacturers do not necessarily use the best grease because of its cost. Because of this, there are instances where you may buy a good lubricated-for-life bearing on the premise that you are getting the best possible lubricant and are really not. In other words, if you know your application, which you should, you should know your grease. With conventional bearings you have more latitude of selection and need not specify your needs directly with the bearing manufacturer. Usually only large users get this privilege. Sealed-for-life bearings are usually only for indoor or protected applications where there is not too much dirt.

Conventional bearings can be lubricated with either oil or grease in most cases depending on the best service available. A great many heavy-duty, high-speed bearings work best with oil. Sealed-for-life bearings are only grease lubricated.

BEARING WEAR

continued from page 41

maximum of 1/4 degree misalignment, or 1/16 inch per foot of shaft length, while loose matched (Code 3) bearings might operate with as much at 1/2 degree misalignment, 1/8 inch per foot of shaft length, provided excessive shaft fits do not remove part of this internal looseness.

Misaligned bearings impose an extra load on the cages to force the balls or rollers up the shoulder and may overheat or cause broken cages. Misalignment also occurs when shafts are too small to prevent deflection under load.

Other Causes of Short Life

When bearings are used in out-of-round housings, they might pinch to cause overload. Early fatigue failure at opposite sides of the outer raceway might result. The latter bearing will also show a brown oxide film on its exterior surface at the fatigued position. This occurs at the position of heaviest load where movement creates frictional heat with resultant oxidation, sometimes called "fretting corrosion" or "friction oxidation." *Finely divided carbon or graphite mixed with light oil applied to bearing seats will decrease this oxide formation and freezing of the bearing.*

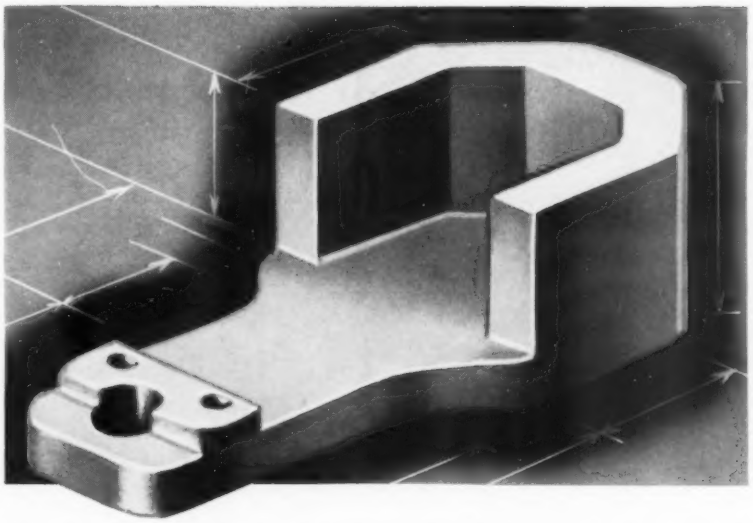
Electrical leakage might occur through bearings from static electricity on line shafts and in direct current generators and motors. This causes sparking at regular intervals around the raceway circumference and is related to the method of winding or position of brushes. Low voltage and amperage during starting may create a negligible discoloration on the raceways but further operation removes metal to form pockets and roughen the surfaces giving them the washboard appearance. High voltage actually burns away metal leaving holes or pits.

Bearings which operate under preload have one ring displaced axially with respect to the other. The actual load is not carried by the ball diameters in contact with both races. Instead, there is a twisting and slipping action at the areas of contact. Simultaneously the balls are pinched and prevented from changing their axis of rotation. This results in wearing of grooves or bands on the balls and in the raceways.

By using duplex pairs of bearings with preloads greater than working

Continued on page 64

economy...speed... in volume production of complex parts...



The photograph shows a bearing support plate for a new line of power tools manufactured and marketed by a large and very capable organization.

The complex nature of the part is apparent at a glance and the cost of machining such a part is evident to the eye of engineer and designer.

It is on parts of this nature that powder metallurgy offers its greatest advantages and its greatest opportunities for the future.

Such parts require most careful designing of the tooling from which they are produced, plus painstaking and tedious effort until the part can finally be produced in volume.

A manufacturer with such requirements naturally turns to Bunting where the necessary persistence until success is achieved is one of the Company's recognized characteristics.

For the unusual, as well as the usual, in bearings, bushings, bars and special parts of cast bronze or sintered metals, see Bunting first.

BUNTING SALES ENGINEERS in the field and a fully staffed Product Engineering Department are at your command without cost or obligation for research or aiding in specification of bearings or parts made of cast bronze or sintered metals for special or unusual applications.

...ask or write for your copy of...

Bunting's "Engineering Handbook on Powder Metallurgy" and Catalog No. 58 listing 2227 sizes of completely finished cast bronze and sintered oil-filled bronze bearings available from stock.

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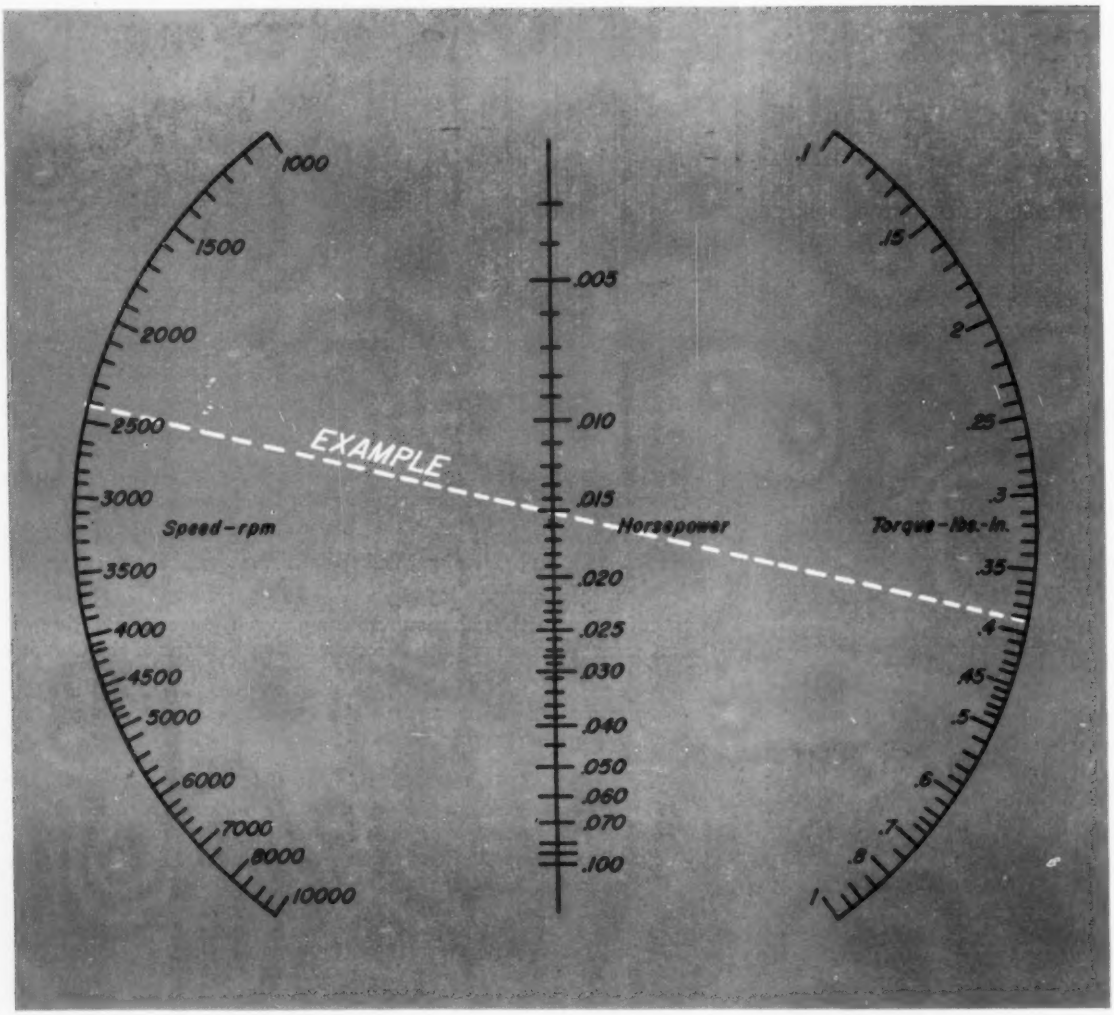
BEARINGS, BUSHINGS, BARS AND SPECIAL PARTS OF CAST BRONZE OR SINTERED METALS.

For more information circle No. 8 on the Reader Service Card

Nomogram of speed-horsepower-torque

Courtesy Barber-Colman Co.



Nomogram is based on the formula:

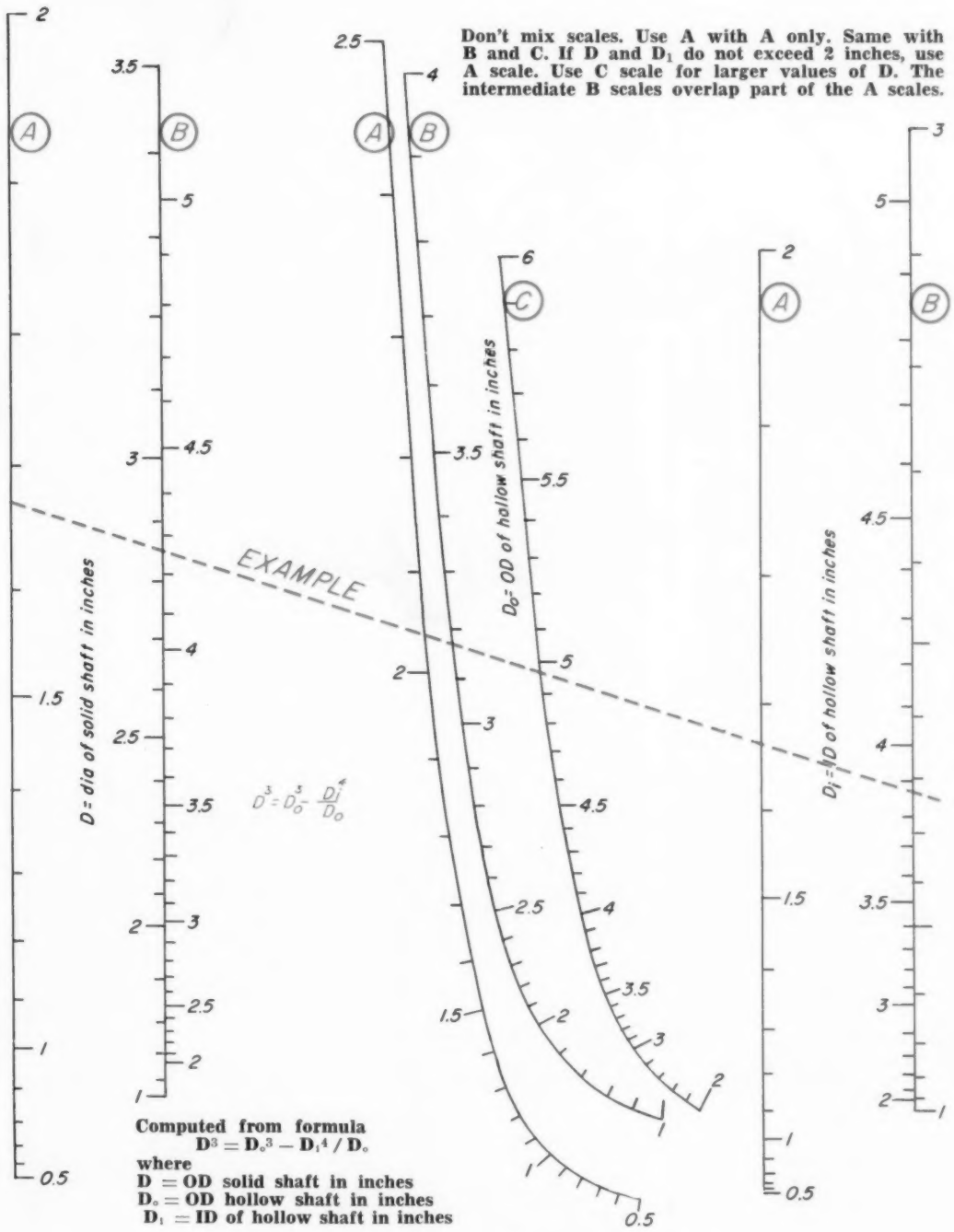
$$hp = 0.00001587 \times \text{speed} \times \text{torque}$$

This is a low scale nomogram. To extend the range of any factor, multiply torque and horsepower by the same factor, or multiply speed and horsepower by the same factor.

For a free reprint, circle No. 200 on the Reader Service Card

Torsional Strength of Hollow Vs. Solid Shafts

By J. N. ARNOLD *Professor, Engineering Graphics, Purdue University*


Don't mix scales. Use A with A only. Same with B and C. If D and D1 do not exceed 2 inches, use A scale. Use C scale for larger values of D. The intermediate B scales overlap part of the A scales.
A
B
C
EXAMPLE
D = dia of solid shaft in inches
Do = OD of hollow shaft in inches
Di = ID of hollow shaft in inches
D3 = Do3 - Di4 / Do
Computed from formula
D3 = Do3 - D14 / Do
where
D = OD solid shaft in inches
Do = OD hollow shaft in inches
D1 = ID of hollow shaft in inches

For a free reprint, circle No. 203 on the Reader Service Card

MEETINGS

MARCH 8-11

American Society of Mechanical Engineers, Gas Turbine Div., engineering conference and exhibit, Netherlands-Hilton Hotel, Cincinnati, Ohio.

MARCH 9-12

American Society of Mechanical Engineers, aviation conference, Statler-Hilton Hotel, Los Angeles, Calif.

MARCH 16-20

American Society for Metals, Western metal show, Pan-Pacific Auditorium, Los Angeles, Calif. Contact: American Society for Metals, 7301 Euclid Ave., Cleveland 3, Ohio.

MARCH 19-20

Society of Automotive Engineers, national production meeting, Sheraton-Cadillac Hotel, Detroit, Mich. Contact: Robert W. Crary, Public Relations Manager, SAE, 485 Lexington Ave., New York 17, N. Y.

MARCH 31-APRIL 3

Society of Automotive Engineers, national aeronautical meeting, aeronautic production forum, and aircraft engineering display, New York, N.Y. Contact: Robert W. Crary, Public Relations Manager, SAE, 485 Lexington Ave., New York 17, N. Y.

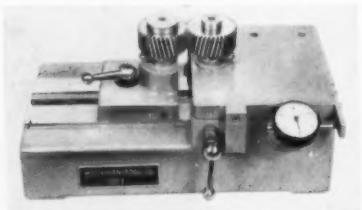
MAY 26-28

Anti-Friction Bearing Manufacturers Association, annual meeting, Drake Hotel, Chicago, Ill. Contact: H. O. Smith, Secretary-Manager, 60 E. 42nd St., New York, N.Y.

MAY 31-JUNE 3

American Gear Manufacturers Association, annual meeting, The Homestead, Hot Springs, Va. Contact: J. C. Sears, Executive Director, One Thomas Circle, Washington 5, D. C.

NEWS *Continued from page 6*



Small fixture checks gear's accuracy

DETROIT, MICH. — Low-cost bench-type gear rolling fixture accurately checks external spur and helical gears. Measuring 7 by 15 by 5-3/4 inches high, the fixture center distance is adjustable from 1-3/4 to 6 inches for checking gears to 10 inches diam. Deflections are shown on the integrated dial in 0.0005 inch increments when the gear being checked is manually rolled against the master gear. Michigan Tool Co. makes the fixture.

Bearing companies bought

ANN ARBOR, MICH. — Strom Steel Ball Co., Erwin, Tenn., and Coolidge Corp., Middletown, Ohio, have been purchased by Hoover Ball and Bearing Co., Ann Arbor, Mich. Board Chairman Clifford H. Simmons said the purchase would add 15% to capacity and should lower manufacturing costs. Management and policies of both purchased companies will not change.

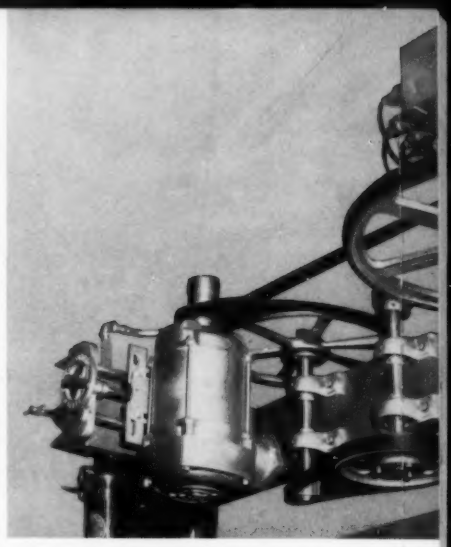
Plant Maintenance Engineering show attendance tops 20,000

CLEVELAND, OHIO—More than 20,000 engineers and plant executives from more than 30 nations attended the Plant Maintenance and Engineering Show in the Cleveland Public Auditorium, January 26 to 29.

Of special interest to power transmission designers and engineers were varied applications of power transmission components in many different types of plant maintenance machinery.

Special gear box service announced

ST. PAUL, MINN. — Gear boxes to order, either from a rough outline of requirements or to a detailed specification will be manufactured by E. B. Sewall Mfg. Co., according to a recent announcement. Or, cooperation at any stage of gear box development is available. The company has been in the gear business twenty years.



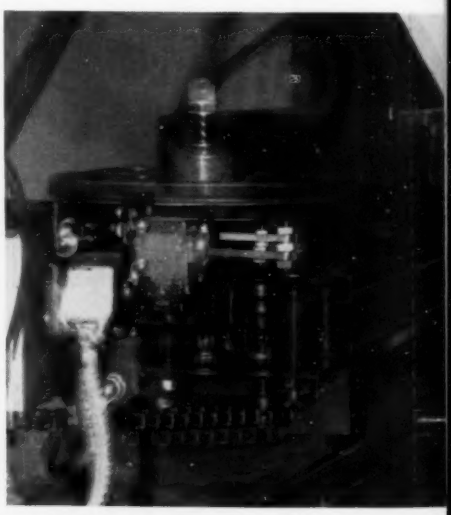
1. BELTS AND PULLEYS work a sprayer

Power drive

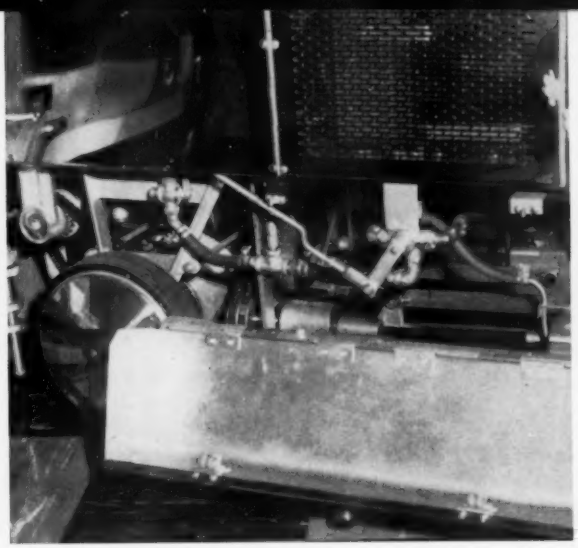
MACHINES USED IN PLANT MAINTENANCE, like others, have good power transmission systems. Here are some of the very interesting setups on machines displayed at the National Plant Maintenance and Engineering Show held in Cleveland January 26-29.

Belt and pulley drive (fig. 1) on an automatic paint spraying machine lets the operator turn the wheel supporting the spray guns by hand for adjustment with no danger of damage to the drive. Gear and pinion are also used to get the needed speed reduction. Made by Binks Mfg. Co., Chicago.

Flexible shafts and right-angle gear



4. ROLLER CHAIN powers this door-opener.



2. **FLEXIBLE SHAFTS** let brushes tilt to extreme angles.

3. **ENGINE TRANSMISSION UNIT** seen in cutaway.

applications at Plant Maintenance-Engineering Show

boxes (fig. 2) drive the brushes on Clarke Floor Machine Co's. floor maintainer. This setup lets the brushes tilt to any angle on their gimbals while supplying constant power. Two travel speeds are given by an interesting clutch, sprocket, and chain layout.

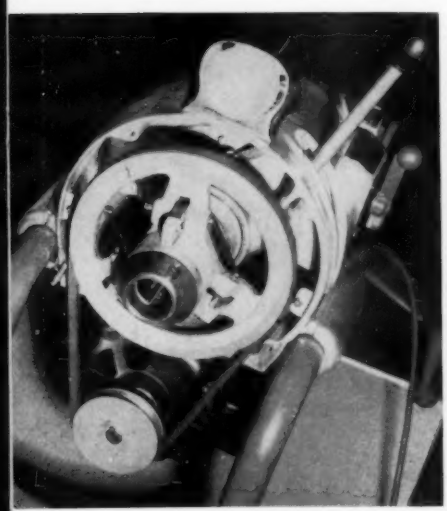
Compact engine-transmission unit for a small tractor includes drive-wheel clutch, planetary gear reducer, differential, accessory-drive clutch, and accessory-drive slip clutch (fig. 3). Planetary gear reducer and accessories are driven through the hollow differential input shaft. Gravely Tractors, Inc., Dunbar, W. Va., makes this one.

Slip clutch, solenoid brake, and a unique roller chain arrangement (fig. 4) open and close garage or similar doors. Roller chain makes a figure 8 to move doors in opposite directions. Designers at Clark Door Co., Inc., Newark, N. J., figured out this system.

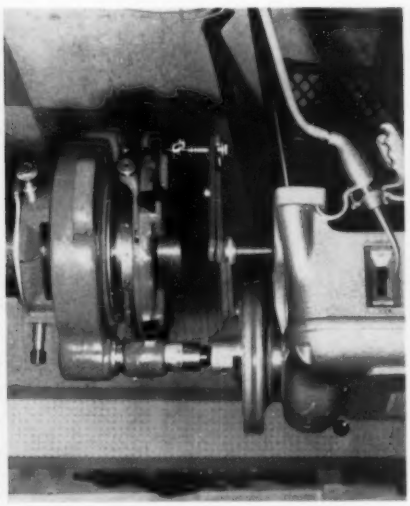
Revolving clutch jaws (fig. 5) drive the spring shafts which rotate and propel the cutter head on this sewer pipe cleaner. Shafts come in 15-ft sections and are easily joined to clean any length pipe. Kollman Mfg. Co., Erie, Pa., makes it.

Automatic overload protection on a pipe threader (fig. 6) is given by a spring-loaded spline clutch. Power from the motor and gear reducer is transmitted through the clutch to a pinion and ring gear. The ring gear encircles the pipe being threaded. Threader is a product of Ridge Tool Co., Elyria, O.

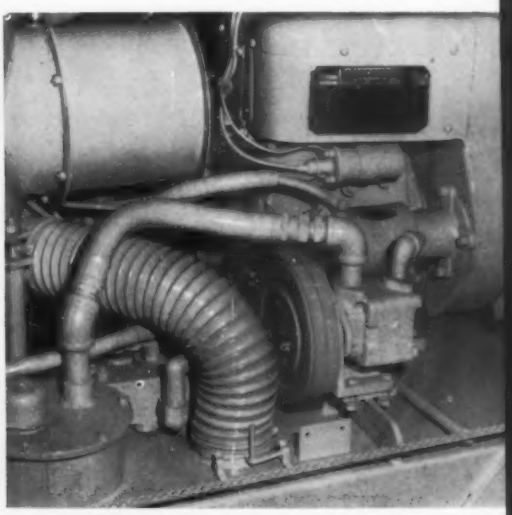
Hydrostatic transmission of power to the drive wheels on an industrial sweeper results in precise control and simple connection to the pivoting drive wheel, (fig. 7). Pump, motor and valves make up the wheel drive. Brushes are both belt and chain driven. Industrial Sweeper Co., South Bend, Ind., is the manufacturer.



5. **CLUTCH JAWS** drive helical spring.



6. **CLUTCH** loosens when cut finishes.



7. **TRANSMISSIONS** are hydro-mechanical.

LITERATURE

on machine drives and related components

Roller bearings

The load zone of a roller bearing is graphically highlighted. Importance of this area is based on the fact that roller bearing life depends much on uniform load-distribution on bearing surfaces.

Roller Bearing Co., of America, West Trenton, New Jersey.

Circle No. 185 on the Reader Service Card

Cam Yoke

Cam follower and cam yoke roller, together with typical applications of these items, are described. Tabular data on widths, stud length, thread, etc., is included. Bulletin 103.

The Smith Bearing Co., Inc., Trenton, New Jersey.

Circle No. 186 on the Reader Service Card

Pancake motor

Construction and design details of an unusual electric motor are given in this interesting story of the drip-proof, pancake motor. A three-dimensional cutaway with callouts is especially interesting. Applications include machine tools, roof ventilating fans, or any cramp-space use. Ratings given are 1 hp to 15 hp. Bulletin 2150.

The Louis Allis Co., Milwaukee, Wisconsin.

Circle No. 187 on the Reader Service Card

Miniature electro-brakes

Sub-miniature electro-magnetic clutches and brakes in models and sizes designed to solve control problems. Drawings, tables, engineering data, specifications, and dimensional data are listed. Catalog 957-A.

Autotronics, Inc., Florissant, Missouri.

Circle No. 188 on the Reader Service Card

Power transmitters

Friction clutches, fluid couplings, torque converters, and marine gears get big play in current edition of Twin Disc's "Production Road". Commemorating 40 years in the industry, the issue tells the story of engine-powered machinery during that time.

Twin Disc Clutch Co., Racine, Wisconsin.

Circle No. 190 on the Reader Service Card

Belts and pulleys

V-pulleys, interchangeable bushings, cast-iron single-groove and two-groove pulleys, pressed steel pulleys, V-belts, V-link belting, and others are shown in 46-page, 6-section, V-Drive Catalog and Engineering Manual.

Maurey Mfg. Corp., Chicago, Illinois.

Circle No. 189 on the Reader Service Card

Actuators

Electro-mechanical control terms are explained in the newly published Electro-Mechanical Actuator Glossary. Included are separate glossaries for rotary and linear actuators.

Hoover Electric Co., Los Angeles, California.

Circle No. 191 on the Reader Service Card

Flexible shafts

Where the engineer and designer should specify flexible shafts is pointed out in this well-illustrated, 4-Page folder. Short and long-coupled shafts are illustrated.

H. S. Watson Co., Emeryville, Cal.

Circle No. 192 on the Reader Service Card

Universal joints

Heavy-duty and extra-heavy-duty universal joints, representative industrial joint arrangements, military standard arrangements, and an attached data sheet for universal joint applications. Catalog 28.

Apex Machine and Tool Co., Dayton, Ohio.

Circle No. 193 on the Reader Service Card

Power drive ideas

Fifty-nine ideas for modernizing your power transmission systems in 1959 detailed in a well illustrated 56-page brochure published for the plant production man as well as the designer and engineer. Among ideas are: Get more accurate variable speeds; Low, low switchgear; Cut DC motor maintenance; An inexpensive way to get constant speeds in lower hp ratings; Clean out your electrical jungle; You can pay for a new pump from power savings.

Allis-Chalmers Mfg. Co., Milwaukee, Wisconsin.

Circle No. 170 on the Reader Service Card

Flexible couplings

Listed in this six-pager are the 14 advantages gained from flexible couplings. An exploded view shows the simplicity of their design. Balls can be removed in two minutes and alignment checked in two easy ways.

The Elliott Co., Springfield, Ohio.

Circle No. 194 on the Reader Service Card

Gearmotors

Descriptions of integral, flexible, and vertical models in 4-page, two-color bulletin. Subjects discussed include "A Size and Type for Every Operation"; Construction; Design; and "Silent Power".

J. D. Christian, Engineers, San Francisco, Cal.

Circle No. 171 on the Reader Service Card

Gear finisher

Details of abrasive, hard-gear finishing make up the interesting contents of this four-page booklet. Outlined are points of machine-operation, gear-crowning, description of the abrasive finishing tool, construction of the machine, and cutting fluids. Bulletin No. 999.

Michigan Tool Co., Detroit, Michigan.

Circle No. 172 on the Reader Service Card

FACTS AT YOUR FINGERTIPS

Your source for copies of any item mentioned in these pages is as close as the Reader Service Cards bound into this issue. Fill them out and mail: no postage needed.

Pillow block
Self-aligning pillow blocks are illustrated as completely assembled, factory-lubricated units. Flange mountings for normal duty, and spherical seat machine units are featured. Dimensional data for self-aligning blocks, flange mountings, and machine units are listed. Bulletin No. 106.

Hoover Ball and Bearing Co., Ann Arbor, Michigan.

Circle No. 173 on the Reader Service Card

Clutches
Great enemy of clutches is heat. Economical clutch-operating life of buses, trucks, planes, and industrial machines is prolonged by use of compressed metallic friction material. How to solve other problems of destructive heat is outlined in 12-page booklet.

The General Metals Powder Co., Akron, Ohio.

Circle No. 180 on the Reader Service Card

Gear reducer
Advantages of the planetary gear reducer are highlighted in 12 informative pages. Included are capacities for: gearing, shafting, bearing, heat, and housing-carrier.

Crichton Co., Johnstown, Penn.

Circle No. 181 on the Reader Service Card

Lubricants
Application of oil and grease, a discussion of "normal" conditions; features on heavy loads, high speeds, and low temperatures add to value of well-written, illustrated data book 57-1.

Lubriplate Div., Fiske Bros. Refining Co., Toledo, Ohio.

Circle No. 182 on the Reader Service Card

Electric motors
Fractional-horsepower, totally enclosed, drip proof, and other types are described in 20 pages of two-color notes and data. Drawing, exploded photo-views, and data are included.

Newman Electric Motors, Inc., Newark, New Jersey.

Circle No. 183 on the Reader Service Card

20% Stronger GEAR TRAINS

with OHIO 20° P.A. Gears

You can add as much as 20% more strength to the gearing in your product, simply by switching to OHIO 20° pressure angle gears.

Ohio offers the only complete line of 20° P.A. gears which have exactly the same face width as the older standard 14½° P.A. type.

Existing designs need not be altered to gain this extra strength because you can substitute a 20° P.A. gear train without changing ratios or center-distances. OHIO's 20° P.A. gears are available for immediate delivery from local stock and they cost no more than 14½° gears.

20° pressure angle gears are stronger because they have a thicker tooth profile with less undercut. For full information on OHIO's 20° pressure angle gears, call your local OHIO distributor or write OHIO GEAR direct.

Get Your Copy Of Ohio Gear's Complete Stock Gear and Speed Reducer Catalog.

Manufacturers of gears and speed reducers - both stock and special

Ohio **OHIO GEAR Company**
1325 East 179th St. • Cleveland 10, Ohio

For more information circle No. 37 on the Reader Service Card

Five methods to determine grease

By DEAN W. CRIDDLE, J. LEONARD DREHER,
California Research Corporation

Slumpability—the ability of grease to flow by gravity—is necessary for proper lubrication of enclosed gears. Without it, gears would turn and form a channel in the grease which would not flow back fast enough for proper lubrication. It is also necessary to keep pumps primed in automatic lubrication systems.

Pumpability is also important. Yield point measurements help to determine if grease will slump and pump.

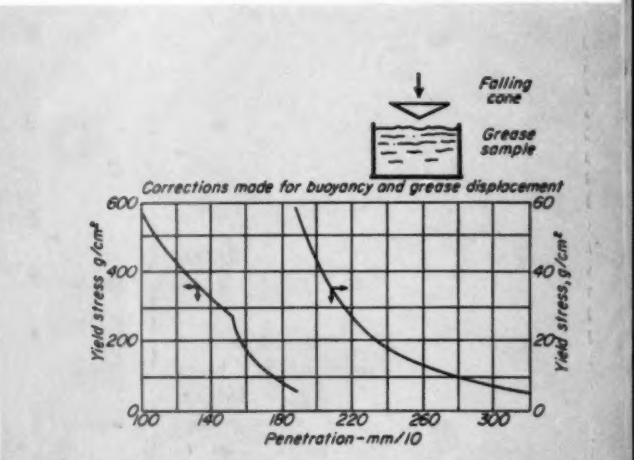


Fig. 1. ASTM cone penetration and the corresponding yield stress.

YIELD POINT is an important property of a lubricating grease. It's a relatively new measure of grease performance, and no standard method of measuring it has been settled. Five methods for determining yield point are described here and compared.

Method One: Cone Penetration

Cone penetration can be used only with greases hard enough to stop the falling cone. Yield stress is calculated from the weight and cross-section of the cone at the grease surface. A buoyancy correction is made by assuming the cone displaced a fluid with the same density as the grease being tested.

Relationship between ASTM penetration and yield point is shown in Figure 1. This method is difficult to interpret because it combines shear and compressive deformation, and shear rate and amount vary.

Method Two: Capillary Viscometer

Minimum pressure needed to cause flow through a capillary is another measure. Data is obtained with a modified ASTM pressure viscometer at a minimum discernible flow rate of 0.0020 cc/sec.

Method Three: Stress-Strain Curves

Yield points are easily calculated from stress-strain, or force-deformation curves. Apparatus used is shown in Figure 2. Curves can be obtained for deformation both in shear and in compression. For tests by both methods, grease is worked 60 strokes in an ASTM worker, prepared samples then aged 10 to 14 days.

Tests are run at 25 ± 1 C. For shear tests, samples are placed between the platens, and the inner cylinder moved at a fixed rate with respect to the outer. Total stress divided by the area of the inner cylinder in contact with the grease then gives shear stress. A typical curve of data gotten this way is shown in Figure 2.

Compression yield points are found with unsupported cylinders of grease. Lengths of the cylinders are at least twice the diameter to minimize end effects. Curves for compression are similar to those for shear. Stress values are higher to produce a given strain.

Method Four: New Flow Equation

A new equation for computing grease flow in capillaries has been developed.

$$\frac{v}{t} = \left(\frac{Pr^4}{8LN_o}\right)\left(\frac{1}{1 + 2.5V + 14.0V^2}\right)C$$

where v = velocity, t = time, P = pressure, r = capillary radius, L = capillary length, N_o = oil viscosity, V = volume percent solids, and C = a function of pressure and yield point.

Method Five: Buckingham Equation

The Buckingham equation is based on an ideal Bing-

ABSTRACTS

of current technical papers

yield points

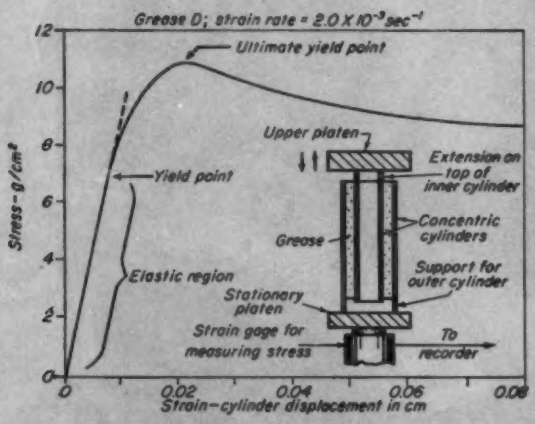


Fig. 2. Stress-strain measurements and the yield point obtained.

Grease sample	Method 1 Cone penetration		Method 2 Capillary viscometer	Method 3 Stress-strain curve		Method 4 Flow equation
	Unworked	Worked		Shear	Compression	
A	4		4	1		2.5
B	12	5	12	3		6
C	25	15	16	5	19	9
D	4	28	27	10	38	22
E	98	57	38	16	59	25
F	140	125	54	28	90	30
G	240	28	—	9	50	—

Stresses given in g/cm²

Fig. 3. Yield point varies considerably with the rate of deformation.

ham solid. However, yield points calculated this way vary depending on whether high or low shear-rate data is fitted by the equation. Therefore, it does not seem that this method is of real significance, and Bingham yield points are not included in the following comparison.

Results Compared

Seven greases, whose compositions and properties are compared in Table 1, are used in this study. Yield points given by the different methods are in Table 2. As shown by Figure 3, rate of deformation affects results. Therefore, there is no single yield point. For a given grease, yield point depends on type of deformation, rate of deformation and the previous working.

Yield points measured under compressive conditions average four times more than those measured under shear. This shows that the penetration method is primarily compressive. Capillary viscometer and flow equation methods appear to give yield points in shear. The viscometer method would undoubtedly give lower values, closer to those of the stress-strain method, if more sensitive flow measuring methods were used.

Obviously, the method used and the test conditions have quite an effect on the yield point. The conclusion is that method and conditions must be stated with the yield point if it is to be valuable.

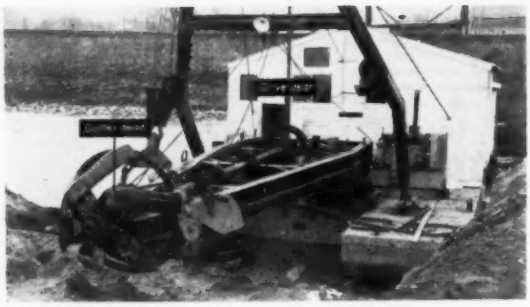
Tests were made on seven greases. Yield stress decreased as deformation rate increased in both shear and compression. This indicates that there is no single yield stress. It is probable that at very low pressures, greases creep as do most solid materials including metal.

Yield stress depends on at least three factors: type of deformation; rate of deformation; and previous work history.

A comparison of shear and compression data obtained by the stress-strain curve method showed that type of deformation is the principle cause of results being different. Yield points measured under compressive conditions averaged four times greater than those under shear.

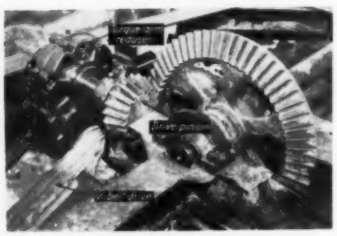
Methods involved using different strain rates vary because of unlike rates of deformation. Different work history will also show different yield points between the worked and unworked greases.

This article is abstracted from a paper presented at the National Lubricating Grease Institute Annual Meeting, Edgewater Beach Hotel, Chicago, October 27-29, 1958.



V-BELT DRIVE is exposed to marine elements.

SLIME AND GRIT are resisted by this open drive unit.



Open drive resists marine contamination

A "C" SECTION V-belt drive powers the cutter-head of a dredge and resists marine contaminants. The belt drive is powered by an 85 hp engine at 1200 rpm. Final speed of the cutter head is 30 rpm reduced through a torque arm speed reducer. All parts are exposed to natural elements of the waterway and operate without downtime or loss of efficiency due to slime and grit.

These contaminants presented the engineers with their most baffling problem in the application of this power transmission unit. Their means of overcoming these specific operational obstacles assured success of the project.

The cutter-head is mounted on a long shaft, powered through a right-angle gear-unit. The entire assembly, including the drive components, pivot up and down with the cutter. Thus, as the cutter is lowered by the crane-rig into the water, the drive units are further exposed to contamination.

This is an excellent example of what can be done with an open drive under the most undesirable conditions. The ever-present, potential contamination of water, slime, grit, and water-dissolved corrosive chemicals made the outcome of this installation seem doubtful. The test has proven that the unit is fulfilling its predictions of efficient operation.

The drive has been operating for a considerable length of time without trouble.

This is an installation of Pentzein, Inc., Omaha, Neb.

SLIP CLUTCHES *continued from page 16*

Duration of one cycle

$t_8 = t_5 + t_6 + t_7 + 0.025 = 0.133$ sec

Maximum cycling rate

$$c_b = \frac{1}{t_8} = 7.5 \text{ cps}$$

Power requirement at maximum cycling rate

$$P_b = \frac{T_L w_b t_7 + T_b w_b t_5}{t_8} = 318 \frac{\text{in-lb}}{\text{sec}}$$

Power dissipation in the slip clutch

$$P_c = \frac{T_b \theta_4}{t_8} = 29 \frac{\text{in-lb}}{\text{sec}}$$

Thus, the high speed clutch is capable of a maximum cycling rate higher by 39 percent than the low speed clutch in this application. In addition, the power requirement is reduced by 24 percent and the power dissipation is cut by a factor of 5½.

Formulae Used

Acceleration of the stationary shaft

$$\gamma = \frac{T_c - T_L}{I_s + I_c}$$

Duration of slip

$$t = \frac{(I_s + I_c)\, w}{T_c - T_L}$$

Angular displacement of the rotating shaft during acceleration period

$\theta_1 = wt$

Angular displacement of the accelerated shaft during acceleration period

$\theta_2 = \frac{1}{2} wt$

Relative displacement between the two shafts during acceleration period

$\theta_3 = \frac{1}{2} wt$

Power requirements during acceleration

$P = T_c w$

Average power dissipation during acceleration period

$$P_c = \frac{T_c \theta_3}{t} = \frac{1}{2} T_c w$$

Barden unveils new plant

Danbury, Conn.—The Barden Corporation has opened a new $2,500,000 plant for the manufacture of instrument precision ball bearings.

The one-floor plant allows production to flow smoothly around a core of service facilities.

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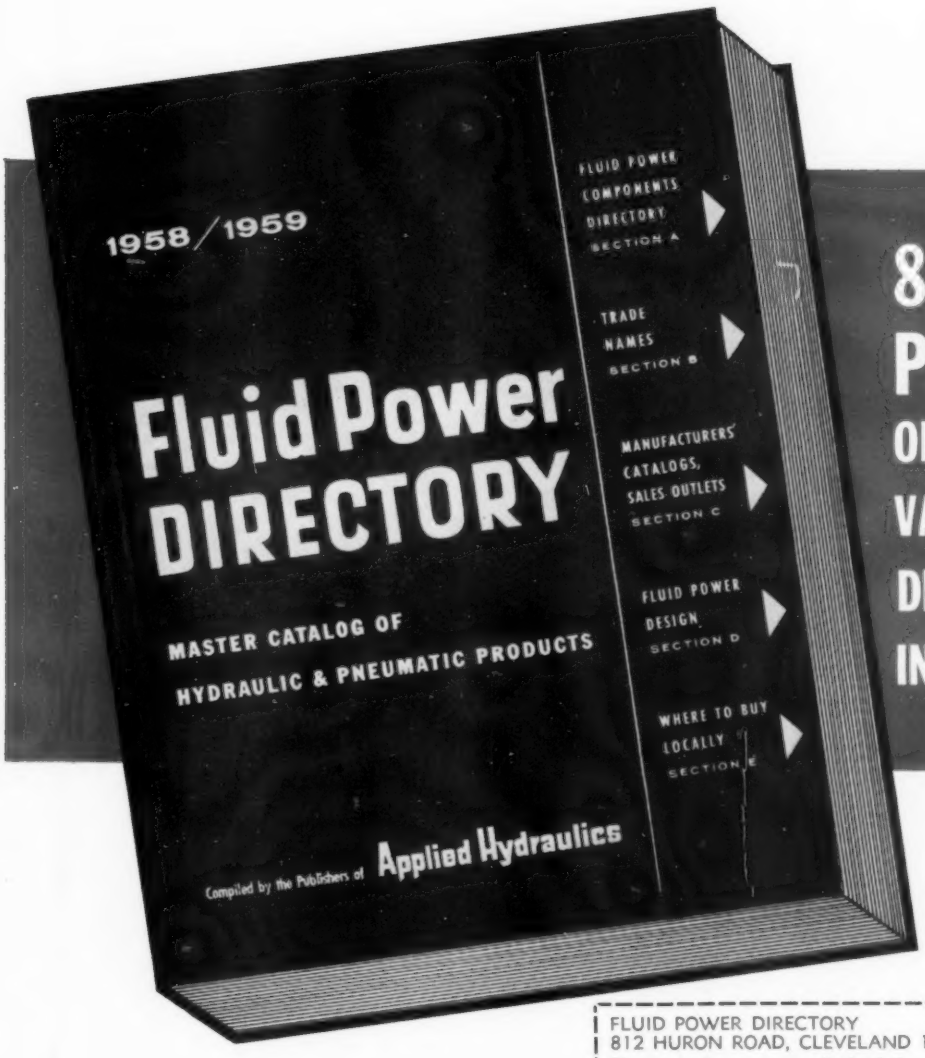
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BOOKS

A Handbook on Torsional Vibration

Compiled by E. J. Nestorides, Senior Research Engineer, the British Internal Combustion Engine Research Association (B.I.C.E.R.A.), Slough, Bucks, England. Published by the Cambridge University Press, 32 East 57 St., New York 22, N.Y.; Price: $19.50.

Especially written and compiled to provide important information for design and development engineers, in a convenient book-form.

The handbook contains much new information, based on accumulated experience and obtained from member-firms of B.I.C.E.R.A., or at the Association's laboratory.

Engine designers will find: recommended formulae, graphical procedures, numerical examples, tabulated values.

Development engineers will find: descriptions of test procedures, details of rigs and instrumentation, precautions against possible sources of error, order of accuracy to be expected of test-results.

Throughout the text, both British and metric units have been included.

1959 SAE Handbook

Published by the Society of Automotive Engineers, Inc., 485 Lexington Ave., New York 17, N.Y.; Price: (Non-Members) $20.00.

Contains all new and revised reports produced by SAE Technical Committees over an 18-month period.

Newly published subjects include: General Engine Nomenclature, Mounting Flanges and Power Take-Off Shafts for Small Engines, Crankcase and Transmission Dipstick Marking, Hydraulic Control Systems for Automatic Transmissions, Lathe Cut Seals.

Technical revisions include: Temperatures for Recommending Gear Lubricants, Starter Pinions and Ring Gear, and others.

Editorial revisions list: Ball and Roller Bearings, Ball and Roller Bearing Locknuts and Washers, Lubrication Fittings, and Engine Test Codes.

This issue includes the 1958 edition, eliminated to allow publication of the 1959 edition on January 1. Subsequent editions will be issued annually on January 1 of each year.

Electricity and Magnetism

By J. Newton, B. Sc., A.M.I.E.E., A. Inst. P., Senior Lecturer in Physics, Northampton Polytechnic, London, England; Published by the Philosophical Library, Inc., 15 East 40 St., New York 16, N.Y.; Price: $10.00.

Balanced and interesting approach to the subject of current electricity, apart from the sections on electrostatics and magnetism which come later.

Chapter headings of probable interest to power transmission designers and engineers are: Direct Current Measurements, Alternating Currents—Machines, Photo-cells, Valves and Cathode Ray Tubes, Electrical Power—Resistance. An extensive index makes this volume a useful reference book.

Steam Turbine Performance and Economics

By Robert L. Bartlett, Design and Application Engineer, with contributions by J. E. Mulder and R. C. Sheldon; Published by The McGraw-Hill Book Co., Inc., 330 West 42 St., New York 36, New York; Price: $12.50.

Contains all important, recent developments in this specialized power transmission field, including advanced steam conditions, largest sizes, and turbine performance.

There is extensive information on performance of single reheat cycles, double reheat cycles, and combined cycles. There is also material on supercritical pressure.

Included are details on the preparation of turbine heat balances and the data needed for those calculations. And for the designer-engineer, performance comparisons are given in graph form.

Industrial Electronics Handbook

Edited by William D. Cockrell; Published by The McGraw-Hill Book Co., Inc., New York. Price: $22.50.

Basic sections bound in a single volume to make up industrial electronics data and information from 102 contributing authors.

Control elements, control circuits, power supplies, mechanical design, instruments, computors—all are covered in the book by fundamental tables, symbols, mathematics, and physical laws.

A final section of the book is of particular interest to designers and engineers. It covers such important things as patent considerations. Also included in this section are other pertinent items such as sources of information.

Principles of Electronics and Electronic Systems

Edited by J. L. Daley, professor, U.S. Naval Academy; Assistant Editor, F. S. Quinn, jr., Commander, U.S. Navy; Authors: G. E. Leydorf, prof. USNA; H. F. Maling, assoc. prof. USNA; J. S. Stoutenburgh, lt. cdr., USN; R. C. Mathis, capt., USAF; E. I. Golding, capt., USAF; H. M. Neustadt, asst. prof., USNA; J. C. Thompson, asst. prof., USNA; P. L. Quinn, instr., USNA. Published by the United States Naval Institute, Annapolis, Md.; Library of Congress Catalog No. 57-13430. Second Edition; Price: $7.50.

The broad basic field of facts needed for effective development of the machinery of the power transmission field must rely heavily on electronics and electronic systems, especially as units themselves become more complicated and complex.

Control and circuit elements are part of the overall picture of power transmission design. This edition includes the basic properties of the circuit elements, and the operation of circuits is analysed. The result is a broad understanding of the principles of electronics.

Engineering Materials Handbook

Editor - in - Chief Charles L. Mantell, Ph.D.; Consulting Engineer; Chairman, Department of Chemical Engineering, Newark College of Engineering, Newark, New Jersey. Published by The McGraw-Hill Book Co., Inc., New York. Price: $21.50.

In line with the needs of the day this most recent edition covers the uncommon metals which are becoming more important economically.

Also, it gathers information needed by everyone who works in the fields of design, structure, machinery, or power transmission. There are tables, characteristics, forms, properties, limitations and advantages.

Engineering Formulas & Tables

Published by Lefax, Sheridan Bldg., Philadelphia 7, Pa.; Price: $9.50.

Includes all types of data needed by engineers and designers in the power transmission field.

Basic formulas, design data, and tables are included.

These sections cover mathematics, measures, materials, gages, screws, mechanics, electricity, magnetism, hydraulics, structural data, and mathematical tables.

It is made in a handy pocket size, convenient for use.

Machinery's Mathematical Tables

Edited by Holbrook L. Horton; The Industrial Press, 93 Worth St., New York 13, New York. Second edition. Price: $3.75.

Contains easily used tables for benefit of machine designers, mechanical engineers and others. These include natural trigonemic functions, logarithms, powers, roots, and reciprocals.

The second edition also includes tables of natural logarithms, logarithms of trigonometric functions, of English and metric weights and measures, English and metric conversion tables, geometric propositions and constructions.

Other data: Tables of dimensions, areas and volumes of plane and solid geometrical figures.

Other tables of prime numbers and factors, and tables of radian measure have also been added. Feature of the second edition is an edged index that makes it possible to locate sections by consulting the flyleaf.

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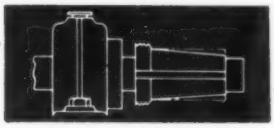

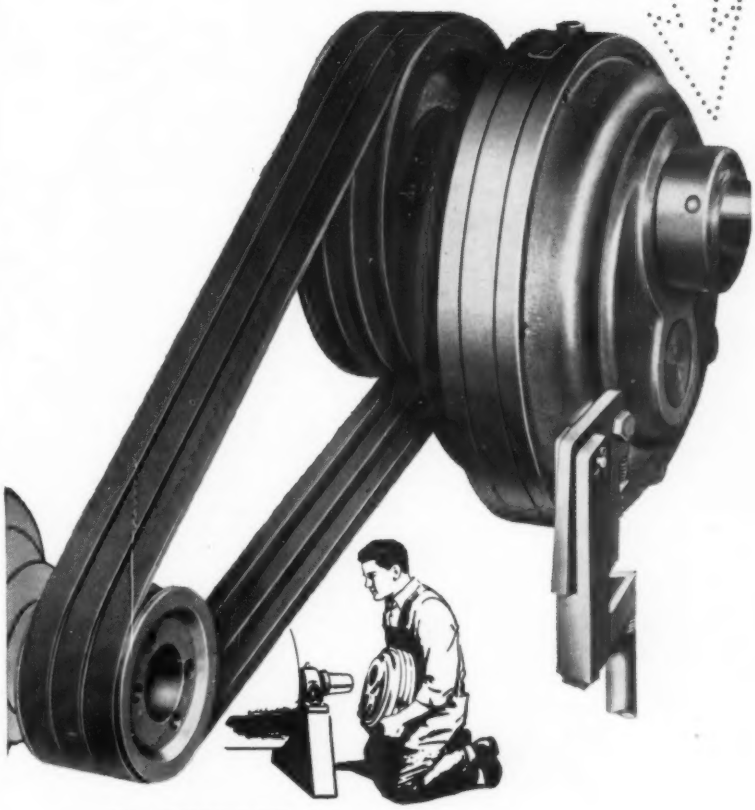

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LETTERS *continued from page 13*

Suggestions

Editor:

Congratulations on your first issue, which was received and looked over with great interest. We like the general style and scope of the publication, including the brevity.

In particular, we like the punched, dotted lines to facilitate tearing out articles of interest, although we seriously object to those articles which are carried over to some other page near the back of the book.

We have one constructive criticism:

> On page 28, the first formula given should have T_p, T_r, and D, all in italics, including the subscripts. On the other hand, KE should remain vertical, or better yet, a single symboy should be created to replace KE. Otherwise, the reader of the formula will think that K is multiplied by E!

May we suggest, also, American Standard B6.5-1954—Letter Symbols for Gear Engineering, available from the American Society of Mechanical Engineers. While this standard deals specifically with symbols for gear engineering, the same general principles apply for many other fields.

GRANGER DAVENPORT
Chief Engineer
Gould and Eberhardt, Inc.
Irvington; New Jersey

• *Sir:*

We are very grateful for any suggestions our readers have to offer. These will be taken under consideration and advisement.

Editor

Editor:

We have noted that you called Mr. Schorsch the manager of our Gear Department. If you will check our approved copy you will find that he is a Gear Proposal Engineer and that Mr. E. C. Denne is the Manager of our Gear Department.

DANIEL L. DAVIS
United Engineering and Foundry Co.
Pittsburgh, Pennsylvania

Sir:

Our deepest apologies to: Mr. Schorsch, Mr. Denne, and the United Engineering and Foundry Co.

Editor

Congratulations!

Editor:

We are very pleased to receive the first issue of POWER TRANSMISSION DESIGN.

It looks, to us, like a very good publication that has been badly needed in this specialized field for a long time.

J. N. GAUS
Mannesmann-Meer, Inc.
Youngstown, Ohio

Editor:

Our sincere thanks for POWER TRANSMISSION DESIGN MAGAZINE. It is an excellent publication and will be very useful to this office.

R. H. COST
Transue & Williams Steel Forging Corp.
Alliance, Ohio

Editor:

I have just had the opportunity of briefly looking over the January 1959 issue of POWER TRANSMISSION DESIGN MAGAZINE.

First, let me thank you for the honor of being on your circulation list.

Second, my congratulations on a magazine that is sorely needed in all industries.

Third, congratulations on the use of perforated pages. I have advocated this scheme for years.

R. L. DRUVA
The Stearns-Roger Mfg. Co.
Denver, Colorado

Editor:

The first issue of Power Transmission Design Magazine has created quite a stir in our company. Our president has put it on the must list for the following:

President
Vice President
Design Engineer
Manager, Development Division
Project Engineer

All of the above individuals are actively engaged in the design of our products and have the authority to specify the components used.

L. C. SHELTON
Ferguson Machine Corp. of Indiana
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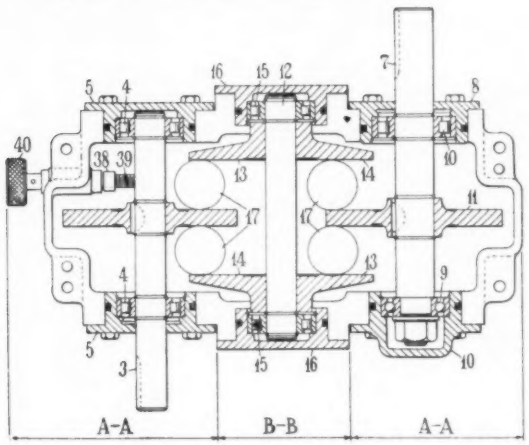
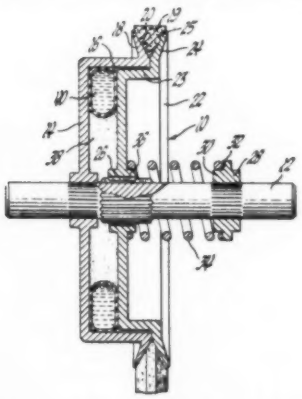
Stepless variable gears



U. S. Patent 2,860,518; William R. Andrews, Burley-in-Wharfedale, England.

A friction disc on the input shaft drives friction discs on an intermediate shaft through balls held between the discs. The intermediate discs drive the output shaft through similar balls and another friction disc. Shaft axes are slightly inclined with respect to each other so that the faces of the friction discs are not parallel. Spring loading and the degree of parallelism causes balls to move with respect to the shaft axes and change the gearing ratio.

Variable-diameter pulley



U. S. Patent 2,851,894; Robert Seyfarth, Mason, Mich., assignor to General Motors Corp., Detroit, Mich.

Automatic decrease in pitch diameter as speed increases in a variable-diameter pulley results when a

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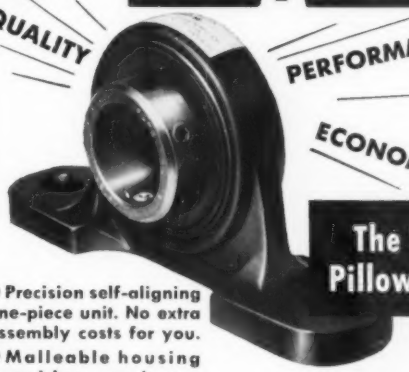

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There's a huge portion of meaty information between the covers of each issue of Power Transmission Design Magazine. Make sure you've gotten the most out of this one before you pass it along. Leaf through it again, particularly noticing things like **New Products,** starting on page 13, and **Useful Literature,** starting on page 52. If you need details on any of them, the Reader Service Cards will bring them to you promptly.

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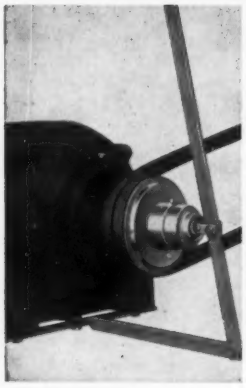
V-Belt Clutch



A New Ball-Bearing V-Belt Clutch for Industrial Drives of 1 to 7½ H.P.

Complete ball-bearing action makes this new "End-Thrust" type Ball-Lok Clutch truly different because of the unique clutching qualities. Easy closing—shockless engagement—instant release—no friction plates or shoes that require replacement or adjustment. Ideal for intermittent high speed clutching. Can't grab! Easy to install.

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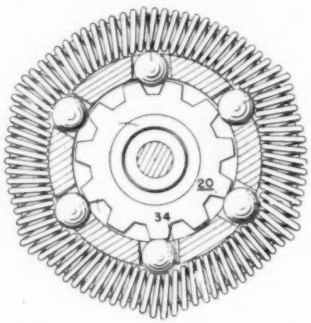
PATENTS *continued*

toroidal tube containing a liquid widens due to centrifugal force. Both pieces of the two-piece pulley are splined to the shaft. One piece is free to slide axially. The other is fixed. A spring, bearing against a collar on the shaft, forces the movable half of the pulley toward the fixed half as centrifugal force decreases.

Ball action slip clutch



U. S. Patent, 2,860,498; Charles N. Crossley, Whittier, Calif., assignor to North American Aviation Inc.

Balls in the outer of two concentric members are driven by engagement in notches of an inner member in this clutch. An even number of balls are used. The inner member has an even number of notches larger than the number of balls. More than one but never all of the balls engage the notches. A double-extension spring wrapped around the outer member holds the balls in the notches as long as torque capacity is not exceeded. When the torque capacity is exceeded the balls are forced out of the notches and the clutch slips.

Variable speed drive

U. S. Patent 2,866,349; John E. Heckethorn, Denver, Colo., assignor to Heckethorn Mfg. and Supply Co., Littleton, Colo.

This is a unit for driving a shaft at either of two speed ratios. It consists of a driver pulley, a driven sheave and a belt that transmits power to a tubular hub fixed to the driven shaft. There is a clutch housing fixed to one of the sheaves and a clutch drum in the housing fixed to the hub. The drum acts at a predetermined speed of one of the sheaves; clutch flanges fixed to the other sheave and drum have a spring mounted between them and the hub. One end of the spring is connected to the remaining sheave and engages when the clutch disengages. This lets the drum shaft drive the driven member through either of two speed ratios. This unit is particularly designed for application on generators for

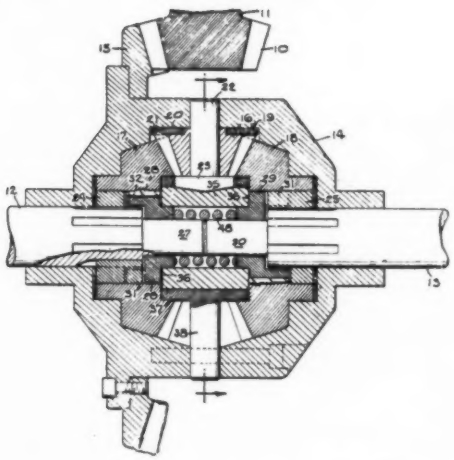
automobiles. The speed ratios are considered those normally found in engines.

Transmission shifter

U. S. Patent 2,866,348; David D. Pettigrew and Edward C. Warrick, Pittsburgh, Pa., to Rockwell Mfg. Co., Pittsburgh, Pa.

For use in a machine tool with a rotary spindle. It consists of a drive member, gear and clutch that alternately connect and dis-connect the drive member from the spindle. It has a manual control with a limited inward and outward stroke that selects desired spindle condition. It also has a locking assembly that prevents operation when the drive member is in rotation.

Differential transmission



U. S. Patent 2,850,922; George I. Welsh, Dixon, Ill.

Cams on the inner ends of the two driven shafts drive followers which can slide axially inward. When one shaft tends to turn faster than the other, the followers move together and drive a collar with a cam surface. A spring between the followers driven by the shafts keeps them apart until a certain minimum speed difference exists between the output shafts to allow differential action. The cam surface on the collar forces friction surfaces against the differential bevel gears and also forces the differential pinions, or spider gears, outward against the inner surface of the differential carrier. Friction facings here brake the differential pinions to keep them from rotating and thus cause pinions, bevel gears and carrier to turn as a unit, and transmit equal torques to both output shafts.

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BEARING WEAR

continued from page 47

load, the balls rotate with only banded areas in contact with the races and smooth quiet operation results. If the spindle is re-adjusted and preload position of the balls is changed, noisy operation will continue until new bands are worn uniformly round or until the bearings are replaced. *Continued vibration may cause roughening of raceways similar in appearance to electrical arcing.*

When excessive thrust loads are applied to single or double row ball bearings, the balls tend to take the internal contact angle caused by the external loads. Compression and deflection of balls and races occur until part of the expected contact ellipse rises above the raceway shoulder. This condition decreases contact area, causes overload and early fatigue. Larger bearings should be used.

When too heavy radial loads are applied, early fatigue occurs. Sometimes these loads will split the rings circumferentially. Larger bearings, roller bearings, double row or duplex pairs of bearings should be used instead to carry the load.

Sound Detection

Bearings sometimes receive the blame for noises since they separate moving parts from stationary parts of machines. However, before the truth can be determined, the sound effect of each component part in the assembly must be individually measured with suitable sound detection equipment.

When bad, bearings on shaft produce increased noise or higher temperature than expected. These bearings should be studied for faulty installation.

PLAIN VS ROLLING CONTACT

continued from page 44

Standardization is still not complete in the bearing industry. However, to permit interchangeability, the bore, outside diameter and width have been agreed upon. In addition, bearings fitting the same shaft or having the same bore are made in four capacities: extra-light; light; medium; and heavy-duty. Similarly, bearings having the same outside diameter are also made in each of these capacities.

To select the proper size and capacity bearing, the manufacturer's instructions must always be adhered to. Each manufacturer supplies a table showing the experimentally-determined radial load which the bearing can support at the best speed, life and other variables. It is, therefore, necessary to convert a particular requirement to a given base. In general, it is done by adjusting the actual load to the tabulated one by means of a series of factors. The factors usually applied are the combined load (radial and thrust) speed, life, race rotation (inner or outer) and shock.

To set up absolute criteria for the proper kind of bearing—plain or rolling-contact—to be used in any particular situation is impractical. It is obvious that hydrostatic bearings are to be used in specialized applications where speeds are real low, loads are relatively high, and where play must be entirely eliminated.

The choice between the hydrodynamic and rolling-contact bearings is not usually so evident. Many factors need to be considered and none of these can always be accounted for in each case.

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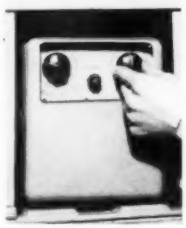
Development by a leading automotive manufacturer of a plant-wide electric motion control program, based on the use of interchangeable power transmission components, provided benefits in lower equipment costs, increased operating efficiency, and reduced maintenance. The purpose of the program was to: (1) reduce peak period air consumption; (2) provide an accurate method of positioning work in automatic machinery; and (3) effect over-all reduction in maintenance costs.

Warner electric brakes and clutches proved to be the key to the program because they provided any desired start-stop cycle, with stepless torque modulation through rheostat tuning in the control panel . . . plus compact design to fit any right-angle or straight-through mounting pattern.



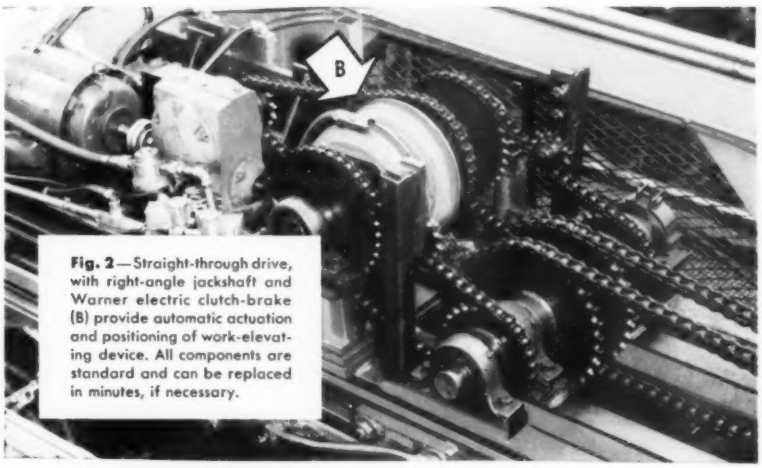


Fig. 2—Straight-through drive, with right-angle jackshaft and Warner electric clutch-brake (B) provide automatic actuation and positioning of work-elevating device. All components are standard and can be replaced in minutes, if necessary.

Simplify to save

All power drives were designed to a set of standards that permitted maximum replacement parts co-ordination and simplified servicing. This has already proved itself by increasing operating efficiency and minimizing equipment downtime. The electric brakes and clutches are a simple method of controlling machinery and synchronizing loading operations through cam-actuated limit switches. Compact control panels, with rheostats for precise adjustment of actuating speeds, are mounted in control centers adjacent to machines. No special shielding is required for brakes or clutches, which operate at duty cycles up to 600 per hour without heating up. No replacement parts have been required in two years of steady operation in this highly automated plant.

Use of Warner electric brakes and clutches eliminated all time lag between actuation and operation of these drives. They require only 30 watts of d-c to energize the magnet and release instantly when de-energized, permitting rapid switching from the brake to clutch, and vice versa. Complex mechanical linkages and piping are eliminated and the only outside connection required is an electrical plug into the control panel. Proof of the modernization program's effectiveness lies in the fact that almost no maintenance has been required for the 150 Warner units used by this plant.

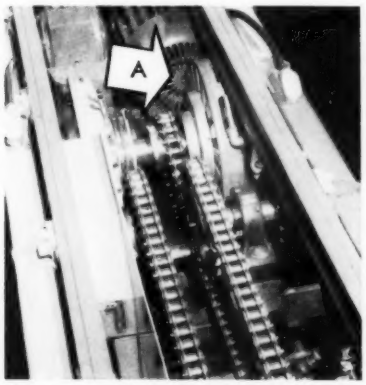


Fig. 1—Compact sprocket drive, with Warner electric brake and clutch (A) mounted on jackshaft, is controlled by limit switches. It provides a high-cycle, one-revolution drive for an automatic loader.

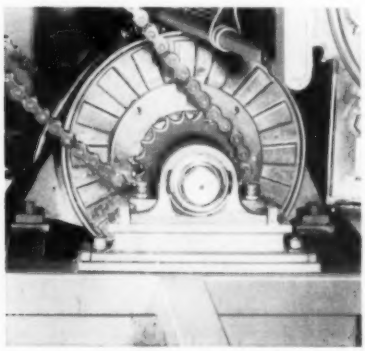


Fig. 3—Automatic loading device is synchronized with production machinery through cam-actuated limit switches. Chain drive to the clutch-brake jackshaft is from a right-angle speed reducer.

Make your power drives profitable

It will cost you less to install electromagnetically controlled power drives than the first year's savings in maintenance and operation. Your Warner distributor can show you exactly what they can do for you that other brakes and clutches can't do. Ask for a desk-top demonstration of Warner *electric motion control*.

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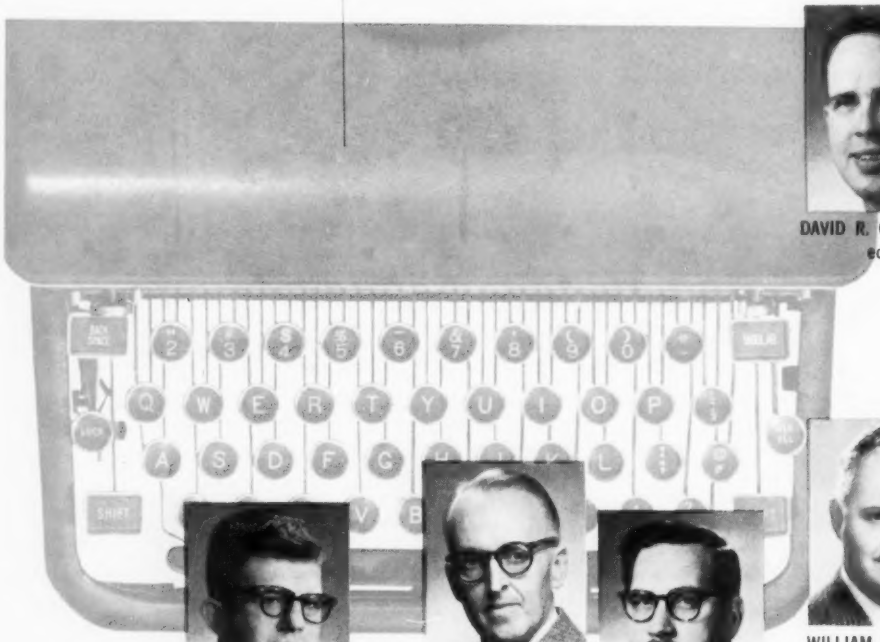

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